STEAM ENGINE INDICATOR AND ITS APPLIANCES

THE STEAM ENGINE INDICATOR AND ITS APPLIANCES.

BEING A COMPREHENSIVE TREATISE FOR THE USE OF CONSTRUCTING, ERECTING AND OPERATING ENGINEERS, SUPERINTENDENTS, MASTER MECHANICS, AND STUDENTS, DESCRIBING IN A CLEAR AND CONCISE MANNER THE PRACTICAL APPLICATION AND USE

—OF THE—

STEAM ENGINE INDICATOR,

WITH MANY ILLUSTRATIONS, RULES, TABLES, AND EXAMPLES FOR OBTAINING THE BEST RESULTS IN THE ECONOMICAL OPERATION OF ALL CLASSES OF

STEAM, GAS AND AMMONIA ENGINES,

TOGETHER WITH ORIGINAL AND CORRECT INFORMATION ON THE ADJUSTMENT OF VALVES AND VALVE MOTION, COMPUTING HORSE POWER OF DIAGRAMS, AND EXTENDED INSTRUCTIONS FOR ATTACHING THE INDICATOR, ITS CORRECT USE, MANAGEMENT AND CARE, DERIVED FROM THE AUTHOR'S PRACTICAL AND PROFESSIONAL EXPERIENCE, EXTENDING OVER MANY YEARS, IN THE CONSTRUCTION AND USE OF THE STEAM ENGINE INDICATOR.

FIFTH EDITION REVISED.

—BY—

WILLIAM HOUGHTALING.

PUBLISHED BY M. E. HOUGHTALING,

BRIDGEPORT, CONN., U. S. A.

1909.

DEDICATION.

To the young Engineers of America, who by thoughtful and careful study; are seeking to make their opportunities and overcome obstacles in the care and management of the Steam Engine; this volume is respectfully inscribed by

THE AUTHOR.

PREFACE.

The preparation of this book has occupied most of the author's spare time for a number of years. Originally the matter was not intended for publication, but the manuscript has grown so large and complete, which consideration, combined with many repeated requests, has induced the writer to publish the matter in book form.

Let every Engineer make his own Indicator book as he proceeds in his study and practice, and it will prove invaluable in after years. The present work has been compiled in this way, from data continually obtained during the author's professional career, extending over a third of a century.

The introduction of algebraical formules have been avoided. These are readily found in the many valuable mechanical Pocket-Books. The writer has endeavored to discuss the principle and use of the Indicator in as plain common sense words as the subject and the English language will admit of.

Special attention has been given to the requirements of the young progressive student in Steam Engineering. The preparation of the following chapters has been a work of pleasure to the author, and if they prove beneficial to his fellow-workmen, he will be amply repaid.

CONTENTS.

	CHAPTER.
Brief History of the Indicator,	I.
Purpose of the Indicator,	II.
Definition of Technical Terms,	III.
Construction of the Steam Engine Indicator,	IV.
Indicator Appliances,	V.
Indicator Appliances, Continued,	VI.
Indicator Reducing Motion,	VII.
Drum Stop and Electrical Attachment,	VIII.
Care and Use of Indicator,	IX.
To Take Diagrams,	X.
Indicator Diagrams,	XI.
Study of Diagrams,	XII.
Lines and Points of the Diagrams,	XIII.
Isothermal Curve,	XIV.
Adiabatic Curve and Point of Cut-off,	XV.
The Foot Pound and Measurement of Diagram,	XVI.
Expansion of Steam,	XVII.
Hyperbolic Logarithms.	XVIII.
Theory of Action of Steam Expansion in Cylinders,	XIX
Reading the Diagrams,	XX.
Different Methods of Computing Water Consumption,	XXI.
Indicator Testing Device,	XXII.
Planimeters,	XXIII.
Comparison Diagrams from Throttling and Cut-off Engines,	XXIV.
The Economy of Expansion,	XXV.
The Point of Cut-off,	XXVI.
Back Pressure and Compression,	XXVII.
Combining the Diagrams from Compound Engines,	XXVIII.
Diagrams From Gas and Oil Engines and Ammonia Compressors,	XXIX.
Making Calorimeter Tests,	XXX.
Miscellaneous Diagrams,	XXXI.
Engine Economy,	XXXII.
Tables,	XXXIII.

CHAPTER I.

BRIEF HISTORY OF THE INDICATOR.

The idea embodied in this important and instructive little instrument, was originated by the celebrated James Watts during the latter part of the last century, at a very early period in the history of the steam engine, and it has since then in its improved forms, materially contributed to the perfection and efficiency of our modern steam engines, not only by enabling the engineer to ascertain the exact values of the forces from which the power is derived, but also by pointing out the precise periods, in relation to the different parts of the stroke, at which these elements of power come into action. The original machine of Watts, consisted simply of a cylinder, about six inches long and one inch in diameter, in which there was a closely fitted piston; and was attached to the engine cylinder by means of a suitable pipe, fitted with a valve to open or close communication between them. A long open coiled spring was used; of which one end was fastened to the piston, and the other to the cover of the indicator cylinder; this spring resisted the pressure of the steam, in one direction, and also the pressure of the atmosphere in the other.

A pointer was connected to the piston, and moved directly with it and served to locate the atmospheric line, or zero; and all motion of the pointer was above or below that point. There was no paper drum; the pointer merely indicating on a scale, the highest and lowest pressure in the engine cylinder, measured from the atmospheric line. An improvement on this instrument was made by adding a flat slide to which a sheet of paper was secured and giving it a coincident motion, on a reduced scale, with the engine piston, by attaching a cord from it to the crosshead or some other moving part of the engine, and returning the same by means of a counterweight. The machine in this improved form, though crude in comparison and less compact in construction, was almost identical in principle of its operation, to the many different instruments now so extensively in use, and it enabled him to ascertain the exact mean effective steam pressure throughout the stroke, and also the proportion which the vacuum in the cylinder, at different parts of the stroke, bore to that in the condenser, in order to determine the dimensions of cylinder required for any given power, as also the relative proportion proper to be given to the steam and exhaust ports, of the slow speed engines of his experiments.

Having attained the objects, he left for succeeding engineers to devise, improve, and put into such a compact and portable form, as to be easily applicable to steam engines of every description; to such an extent as we find the modern indicator of to-day.

One of the first to improve on the instrument of Watts, and nearly one-half a century later, was Wm. Macnaught of Glasgow, who constructed an instrument represented at one-quarter actual size in elevation, Fig. 1, and in plan, Fig. 2.

Unlike the indicators of to-day, that have a parallel and multiplied movement of the pencil as compared with the piston; in this instrument the movement of both are coincident,

that is, whatever motion is imparted to the piston by the steam, the pencil moves precisely the same distance either way. As constructed in this way, the spiral spring opposing the force of the steam against the piston, is liable to disarrangement, and unevenness; on account of the greater length of spring necessary to obtain cards of a convenient height for computation.

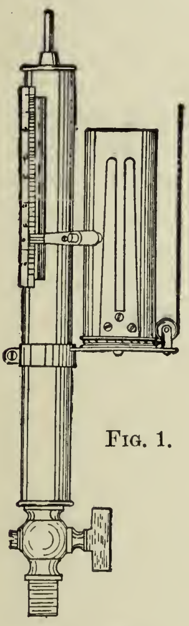


Fig. 1.

Its consequent general adoption has led to many very important, and decided improvements in the construction of the instrument, and very materially aided in elevating the standard of duty of steam engines, and also demonstrated the economy resulting from a liberal use of the expansive power of steam.

From a glance at Figure 1, it will be perceived that the Indicator is neither more nor less than a small single acting steam engine with the addition of a spiral spring on the opposite side of the steam piston, to resist the force of the steam.

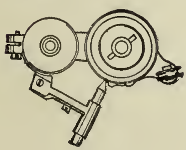


Fig. 2.

The many treatises at the present day on the steam engine indicator, leaves little to be said in reference to the actual manipulation of the instrument in practice: and we now find almost all of the Stationary, Locomotive, Marine, and other engineers imbued with the necessity of understanding the working of the indicator in all its details, not only in the interest of their employers, but particularly for their own personal benefit, in acquiring a knowledge of the intricate working of steam, or other operating forces, of their engines, in all their various details, and which becomes necessary to enable them to reach a place in the higher ranks of engineering science.

The principal parts of an indicator, of the most complete construction, consists of an outside cylinder, or body, inside of which is secured an inner or working cylinder; with a nicely fitting piston, of exactly one-half of a square inch in area, equal to about .7978 of an inch in diameter, working therein.

The piston is so closely fitted to the working cylinder of the indicator, that at the pressures of from two to three hundred pounds per square inch, there is but a very small amount of steam that can pass by or through it; but all indicators are subject to a certain amount of leakage of the piston, and for which ample means are always provided for its escape; in order to avoid any unnecessary back pressure on the side of the piston opposite the steam pressure.

Where the indicator is to be used for obtaining cards, from pressures ranging from four to six hundred pounds per square inch, it is advisable to use a reduced size of working cylinder and piston; the area of which, shall only be one-fourth of a square inch; equal to about .5641 of an inch in diameter, thus avoiding the use of indicator springs of a very high tension. The different sizes of the working cylinders, and their pistons, are made interchangeable in the indicator, so that either can be removed at any time, and the other substituted.

One end of the piston rod is connected to the piston, by a ball connected joint, and the opposite end to some part of a pencil mechanism, for producing a so-called parallel, or straight line movement of the pencil, up and down, in reference to the paper drum.

The parallel motion is another beautiful and ingenious invention of Watts, as he applied it to steam engines, for the purpose of guiding the piston rod, back and forth in a straight line; and without the intervention of any guides, or other means, for the purpose; thereby eliminating friction to a great extent; with smooth working, and providing a mechanism that has since been used in various modified forms, and adapted, to

a certain extent, to different kinds of machinery where a motion of this principle is desired.

The parallel motion, as applied to his engines, was considered by Watts, notwithstanding his other inventions, to be the one which gave him the greatest satisfaction and pride, as it performed its mission with the best practical results; even though not absolutely mathematically correct in some respects. In most of our modern indicators this principle of mechanism, for their pencil movement has been almost universally adopted in its various and possible modified forms, and constant efforts in this direction are being made, with a view of eliminating the slight inaccuracies that are known to exist in the original parallel motion; particularly in the constant varying ratio of the movement, that exists between the piston and the extreme end of the system or the point at which the pencil is located.

The parallel or straight line movement of the pencil is usually produced by a system of levers and links of definite lengths, and pivoted in such a manner that by any movement of the system, up or down, the end of the lever carrying the pencil is always expected to move in a straight line, and also in some exact ratio to the movement of the indicator piston.

CHAPTER II

PURPOSE OF THE INDICATOR.

The Steam Engine Indicator is an instrument originally designed for recording the varying pressure of steam in an engine cylinder; at any, and all points during the revolution of the engine, and has subsequently been applied under many, and various other circumstances, wherever the record, and measure of an irregular pressure has been desired.

The production of this record is the result of two motions, and is traced by the indicator pencil upon paper that is secured to a light cylinder, called the paper drum.

To this drum is imparted a rotary oscillating motion upon its axis, at right angles to the motion of the pencil; such motion being derived from the cross head, or any part of the engine having a movement coincident with it.

The motion of the pencil is produced by the varying pressures of steam acting against the indicator piston; in opposition to the strength of a spring of known tension. Consequently, an indicator properly attached too, and communicating with the interior of the cylinder of a steam engine in operation, and the drum given a motion (on a reduced scale) corresponding to that of the engine piston, will, (on bringing the pencil in contact with the paper upon the drum, during its oscillation), trace the outline of an irregular figure upon the paper, which is usually known as, and called an Indicator Diagram,

an example of which is represented in Fig. 3, C. D. E. F. B. B. and shows the varying pressure acting against *one side only* of

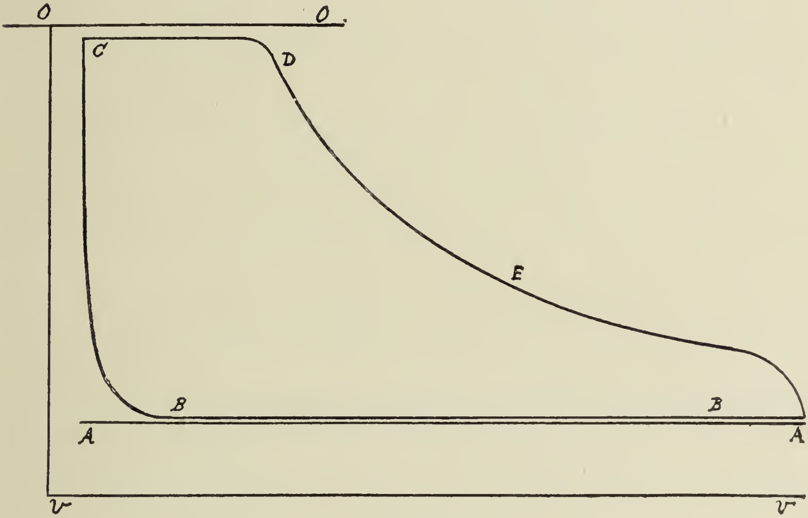


Fig. 3.

the piston, during a revolution of the engine; and such pressures can be properly located, and correctly measured.

The upper line C. D. E. F. of the diagram represents the force of the steam impelling the piston during its forward stroke; while the back pressure line B. B. shows its retardation on the return stroke; the average height of each being measured, (by the scale of the spring), from the atmospheric line A. A.

The difference between their average height, represents the Mean Effective Pressure, usually designated the M. E. P. of the diagram.

The atmospheric line A. A, is also drawn by the indicator but at a time when steam communication between it, and the engine cylinder is closed; and consequently subjecting both sides of the Indicator Piston to atmospheric pressure only.

To show the pressure on the other side of the engine piston, another diagram must be taken from the opposite end of the cylinder.

In most calculations of the diagrams, it becomes desirable to draw, by hand, additional lines, from and by which, the actual lines may be compared.

First, the straight line V. V. is drawn to represent the absolute vacuum, or absence of all pressure.

Second, the line V. O. represents the clearance volume, and is drawn at right angles, (or perpendicular), to the atmospheric line.

Third, the line O. O. is drawn to show the full boiler pressure in order that it may be seen on the diagram how near that pressure has been realized.

The different lines, and events of the stroke, as shown by the diagram during a revolution of the engine, and the name by which they are designated will be noted, and explained in a later chapter.

Diagrams taken from different engines, and under varying conditions, will present themselves in an almost endless variety of forms, depending entirely upon circumstances connected with the operation.

The earlier forms of the instrument, (although the same in principle), were crudely constructed, the moving parts exceedingly heavy; inducing vibrations, sufficient to vitiate, and distort the diagrams; also causing more or less tardiness at the different events of the stroke, such as admission of steam, point of cut-off, and also compression; because of the heavy parts being unable to respond so promptly to change of pressure; thereby making them unreliable, and imperfect in their action, to an extent depending upon the weight of the pencil mechanism, rapidity of rotative speed, and also suddenness of change of pressure; and therefore preventing their being successfully adaptable, *only* to engines of very low rotative speeds.

The prevalence of many high speed engines in use at the present time, renders the use of the old type of indicators almost useless, and wholly unsuited for high speed, where any reliable results are expected, as many details which gave little trouble at low speed, seriously affect the results at the higher speeds.

The present improved construction of some of the various modern indicators, in which perfection is attained as near as can be expected, consists principally in superior designs, simplified construction, a better adaptation of the pencil mechanism, also finer adjustment, convenience of manipulation, and especially extreme lightness of the moving parts, thereby practically eliminating the effects of inertia, and momentum of these parts.

These qualities in any Indicator are absolutely indispensable in order to secure accurate and reliable results from high speed engines.

The information that may be derived from a careful and attentive application of the Indicator to engines of all descriptions, is almost incalculable to the engineer; because many facts are accurately determined by its use, that cannot be obtained in any other way, with any great degree of satisfaction or correctness; consequently its use has enabled the engineer to discover many unforseen, and necessarily unknown defects existing in the engine, that have formerly been veiled in mystery, and at the present time, its value is so universally recognized, and relied upon, that most manufacturers of high grade engines, make provision for its application; and do not consider their engines complete until the valves have been correctly adjusted by the use, and assistance of the indicator, and set in such manner, as that the maximum efficiency of the engine shall have been attained.

CHAPTER III.

DEFINITIONS OF TECHNICAL TERMS.

The many terms, generally used in connection with the study of steam engineering, are all measurable; each with reference to some established *unit*, and by which they are clearly defined, and their correctness recognized.

Some of these are indispensable to the engineer, a few of which are briefly described, and explained as follows:

The Unit of Work is equal in amount to the power required to lift one pound, one foot high, and is called the Foot Pound.

The Unit of Heat, or Thermal unit, is the quantity of heat necessary to raise the temperature of one pound of water, one degree, or from 39° to 40° Fah.

Also, one unit of heat is equal to 772 foot pounds or units of work.

The Sensible Heat of any body, as Air, Water, or Steam, is the heat that is sensible to our touch, and in extent is as shown, and measured by the thermometer.

Latent Heat of steam is the quantity of heat, (expressed in heat units), required to vaporize *water*, that has previously been heated to a temperature equal to the resulting steam, of vaporization.

Specific Heat is the quantity of heat measured in thermal units, necessary to raise one unit of weight of the substance, through one degree of temperature.

Saturated, or Dry Steam, is steam confined under pressure in contact with the water from which it is formed, and contains just sufficient heat to maintain the water in a state of steam, and will vary in pressure, and density corresponding to varying temperatures.

When saturated steam suffers any loss of heat, a condensation of some of the steam also takes place.

Superheated Steam is steam containing an excess of heat.

If to saturated steam more heat be added, its temperature will increase, and the steam is said to be superheated, because its temperature will be greater than that corresponding to *saturated* steam of the same pressure. This excess may be parted with, without condensation.

Horse Power (H. P.) is the standard used for measuring the power of a steam engine, and is equal to lifting 33,000 pounds one foot high in one minute of time, or 550 pounds in one second, or any equivalent thereof in opposition to the force of gravity.

Indicated Horse Power, (I. H. P.) is the horse power of an engine as computed from the Indicator Diagram.

If the mean area of the piston in square inches be multiplied by the mean effective pressure in pounds per square inch exerted against it, and also by its speed in feet per minute; this product on being divided by 33,000, will be the indicated horse power of the engine.

Net Horse Power is the indicated horse power of an engine *less* the horse power which is consumed in overcoming its own friction.

Boiler Pressure is the pressure above atmosphere, or the pressure as shown by a correct steam gauge.

Initial Pressure is the pressure in the cylinder acting against the piston, at or near the commencement of the stroke of the engine.

Absolute Pressure is the pressure of the steam calculated from absolute vacuum.

It is the pressure as shown by a steam gauge, with the pressure or weight of the atmosphere added thereto.

Terminal Pressure is the pressure above the line of absolute vacuum that would exist in an engine cylinder, provided the release of the steam had not taken place until the end of the stroke had been reached.

Usually the release happens earlier, and in such case its position may be located by continuing the expansion curve by hand from any point of release, to the end of the diagram.

The terminal pressure must always be measured from the vacuum line; consequently it is the absolute terminal pressure.

Mean Effective Pressure (M. E. P.) is the *average* of all the varying pressures at different parts of the stroke, exerted against the engine piston to impel it forward; *less* all the pressure that acts on the opposite side of the piston to retard its progress.

Back Pressure Line, in a *non-condensing* engine, represents the loss that occurs, from the retardation of the escaping steam; due to atmospheric or other pressure

It is indicated on the diagram by its height above the atmospheric line, and is expressed in pounds per square inch.

On the diagram of a condensing engine it is indicated by its height above a line drawn by hand to represent the absolute or perfect vacuum.

Total Back Pressure is represented on the diagram by its height above the line of Absolute Vacuum.

Initial Expansion is the fall of pressure along the steam line; which often happens in an engine cylinder, between initial pressure, and the point at cut-off.

Compression is a result caused by the action of the piston in compressing into the clearance space, all steam remaining in the cylinder after the exhaust valve closes.

Clearance is all of the space or waste room between the piston at the end of its stroke, and the face of the valve.

Its volume or amount is usually expressed in its per centage of the piston displacement.

Piston Displacement is the distance passed over by the piston in traversing a single stroke.

Its volume is equal to the area of the piston in square inches, multiplied by the length of stroke, in inches, the product is the volume of displacement in cubic inches.

Wire Drawing is a term sometimes applied to the action of steam, and arises in consequence of very restricted steam passages, also dilatory valve motion, thereby reducing more or less, the pressure in the engine cylinders, and usually considered to be a loss in the matter of economy.

Valve Lap is the excess of length of the valve at each end; (when at the middle of its stroke) over the extreme outer edge of the steam ports; and is designed to serve as a cut-off valve within certain limits.

Valve Lead is the amount of opening of the steam port, (which is regulated by the valve) for the admission of steam to the cylinder, *just as* the piston arrives at the corresponding extreme end of the stroke; the entering steam thereby serving as a cushion for the reciprocating parts of the engine.

Water or Steam Consumption is the amount of steam accounted for by the indicator, per horse power per hour; and is a measure of the economy of the engine.

CHAPTER IV.

CONSTRUCTION OF THE INDICATOR.

The principal difference in the construction of the various indicators on the market to-day lies in the pencil movement mechanism. The main objects sought in the design of this mechanism are as follows: 1st, the nearest possible approach to straight line vertical movement of pencil parallel to the axis of the steam cylinder. 2nd, constant ratio of movement, pencil to piston. 3rd, lightness of moving parts, thereby reducing the momentum of these parts to a minimum. 4th, accessibility of the parts and convenience in handling same to take diagrams.

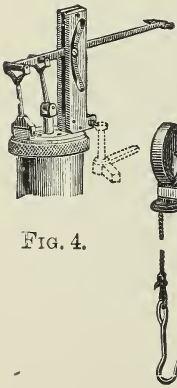


Fig. 4.

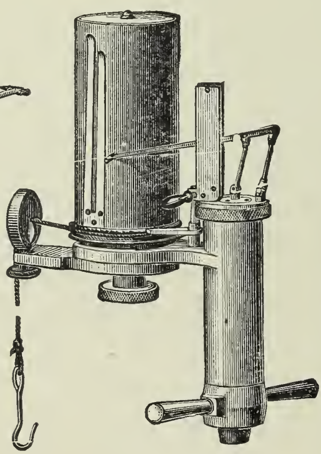


Fig. 5.

For our intent and purpose, it is not necessary to describe more than one of the various instruments in the market to day, and we have in preference to all others, selected the well known improved new style Tabor Indicator, as shown in Figs. 4, 5 and 6 as being the instrument more nearly fulfilling all the con-

ditions and requirements named; and which are absolutely necessary for the production of the most accurate results in Indicator practice, including lightness of moving parts; thereby reducing momentum; and also for ease and convenience to the operator in handling the instrument while taking diagrams.

In order that the particular advantages of this instrument may be thoroughly understood, it is necessary that a brief description of its construction, and arrangement of parts, should be given here.

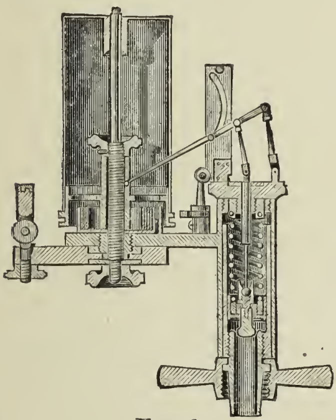


FIG. 6.

The principal and most important peculiarity of this instrument from all others, lies in the method employed to communicate a straight line motion to the pencil, and at the same time of producing an exactly equal ratio of movement between the piston of the Indicator and the pencil; both of which are perfectly accomplished by means of a steel plate, through which there is a slot (as shown, full size, in Fig. 7), of such contour, or shape, as to exactly counteract the tendency to a radial movement of the pencil bar; this slotted plate is attached in an upright position on the swivel plate of the instrument, and upon which the whole pencil mechanism is self-contained. The swivel plate is in turn secured to the cylinder cover of the instrument; said cover serving as a center, upon which the entire pencil movement can be revolved in either direction, until it comes in contact with the paper drum. In order to make this slot in the plate available for the purpose, a small stud is secured on the pencil bar, upon which there is mounted a hardened steel roller, fitted so as to travel in the slot from one extreme to the other; and in doing this,

the pencil is caused to move in a straight line up and down the paper drum, in an exact ratio, throughout any and all parts of the movement, of exactly five times the distance moved by the indicator piston.

The radial slot in the plate is an irregular curve and deviates slightly from a true circle; the irregularities existing in the curve, just compensating for the error that would occur in case a radial link were substituted, and corresponding in length to the average radius of the slot; where one end of said link is pivoted to a stationary point and the other to some part of the pencil mechanism; a method now employed in a number of the well known indicators at the present time, and the results are, that the lines made by the pencil, which are supposed to be straight, are not so, nor are the ratios of piston and pencil equal.


Fig. 7.

A radial link, the end of which always moves in a true circle, is therefore not a successful substitute for accuracy, as compared with the irregular curve in the slot plate of the Tabor Indicator, as said slot which guides and controls every movement of the pencil being so formed as to give the most accurate results, in every position the pencil bar may assume.

There is another plate of exactly the same outline, as the slot plate (but without slot) and secured coincident with it; the roll stud in the pencil bar projecting far enough through it to make contact with said plate, and serves to receive any pressure that may occur when the pencil is brought in contact with the paper drum in the act of taking diagrams; and it also prevents the torsional strain that would otherwise come on the center and back links, thereby reducing the friction from this cause to a minimum.

These plates are provided with a projecting part on their lower ends, which are drilled and tapped to provide for a screw passing through, to regulate the pressure of the pencil upon the paper drum, said screw coming in contact with a standard secured to the body of the instrument. A small upright projection on the swivel plate, serves as a fulcrum for the lower end of the back link, the upper end being connected to one end of the pencil bar.

The back and center links are made of a fine grade of steel as also the pencil bar, which is hardened and carefully drawn to a spring temper, highly polished and all given a blue finish.

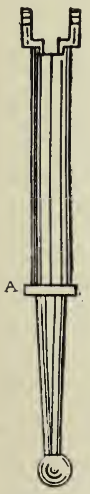


FIG. 8.

The piston rod is of steel and made hollow in order to reduce weight, its upper end is connected to the center link, and the lower is made solid and terminates in a ball; this ball is provided with a universal cap and socket and to which, in turn, the indicator piston is attached by means of a thumb nut.

The piston rod is formed of three pieces; the body, shank and ball; and upon the shank there is formed a collar as shown at A, Fig. 8, full size, a little larger in diameter than the body, and serves as a safety stop for the pencil movement, by coming in contact with the under side of the cylinder cover just before the pencil bar reaches its extreme height, thereby preventing any further movement of the piston rod, which would be likely to result in injury to the pencil movement in case of breakage of a spring when in use, or by a mistake of using too light a spring for the steam pressure.

The piston is made very light, of a hard bronze metal, truly turned, grooved on its periphery, to act as a water packing, and lapped perfectly round and straight on its face to an exact size.

On the projecting arm of the instrument is secured a steel center stud, extending to the top of the paper drum, and around

which all moving parts of the drum mechanism oscillate. The spring case has a threaded hub and is permanently secured to the stud, and rests directly on the top surface of the arm and is secured thereto by a nut underneath.

A flanged disc, or pulley, which carries the paper drum has a projecting hub on both top and bottom, which insures a long and accurately fitting bearing on the stud, working almost frictionless.

There is a hook secured to the bottom hub, which engages one end of a plain flat spiral spring, while the other end of the spring connects to a similar hook in the spring case. A part formed on the disc, is made to come in contact with a stop secured to the projecting arm of the instrument and serves to always locate it in a positive position, when alone under the tension of the spring.

The lower hub of the disc rests directly on the spring case, while the opposite hub is in contact with a knurled thumb nut, screwed and pinned to the central stud, just sufficient to give a slight amount of end motion to the disc.

This thumb nut also serves as a convenient means of regulating the tension of the spring, as by loosening the nut that secures the spring case to the arm of the instrument, said thumb nut can be turned in either direction until the desired tension is obtained, and then tightening the nut.

The lower part of the disc is formed with a groove, wide enough to receive about two turns of the cord; one end of the cord being made fast to, and encircling the groove, and the other attached to a pantograph, pendulous lever, or some sort of a reducing gear that has a movement generally derived from the motion of the engine crosshead, and which causes the disc to rotate in one direction; the motion in the opposite direction is accomplished by the retraction of the spring connecting the disc with the spring case, and its oscillations are thereby made coincident with the movement of the engine piston.

The drum upon which the paper is placed in taking cards is a very light cylindrical tube, mounted on the disc, and moves in unison with it. It has a guide permanently secured on the inside, and which fits in corresponding recesses in the disc, in order to serve as a carrier, and also to locate it in its proper position in reference to the pencil bar, for either right or left hand indicators. The top is closed and fitted with a sleeve that provides a bearing in contact with the central stud, and which serves as a guide for it, and prevents any irregular motion at that point. A clip is attached for securely holding the paper, one leg of which is made shorter than the other to facilitate the matter of adjusting the paper upon the drum.

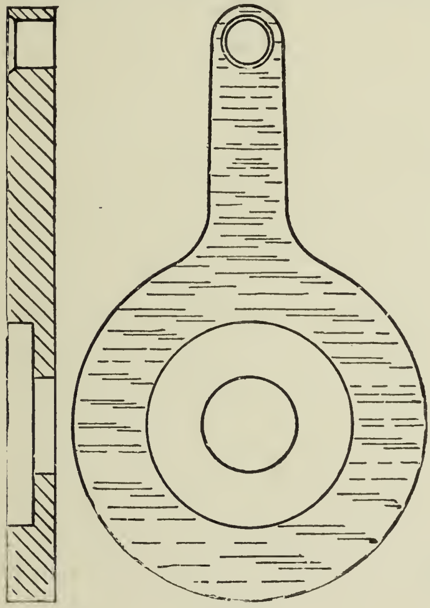


Fig. 9.

One end of a plate of suitable outline, is shown in full size Fig. 9, and called the cord guide-base, is secured to the under side of the arm of the instrument by a nut for that purpose, and said plate can be turned in any desired position parallel to it. The other end supports the cord-guide, which consist of a small grooved pulley mounted on a pin within the periphery of a circular disc, both being encircled by a clamp having a threaded stem, which projects through the plate, and is secured thereto in any desired position by a small thumb-nut, and the combination of the adjustment enables the cord to be correctly guided around the pulley and on to the flanged disc from any direction.

The cock tube is securely screwed in the body of the instrument at the bottom, and the connection which secures the indicator to the cock is made with a single thread, and secures correct attachment at once without the annoyance of different trials, as is often necessary with connections made with two threads.

The indicator cock has a stop, which limits its range in either direction to full open or closed, and also has holes provided for the release of all steam that may remain between the indicator piston and cock, after operating. In taking cards where two indicators are being used, one on each end of the steam cylinder, the straight cocks shown in Fig. 14, Chapter 5, are all that are necessary for the production of cards from each on separate sheets but in the case of only one indicator being used there is considerable trouble and annoyance, as well as time lost to the operator, in changing the instrument from one end of the cylinder to the other. All this may be entirely obviated by the use of the three-way cock shown in Fig. 10, this cock being interposed about midway of the steam cylinder in a line of pipe connecting the two straight cocks at the ends of the cylinder. This three-way cock furnishes an easy and convenient means of taking cards from each end of the cylinder, and within a few seconds of each other; also on the same sheet of paper; and therefore admits at once of a ready comparison between the cards. At all other times than applying the indicator, the cocks near the cylinder should be kept closed.

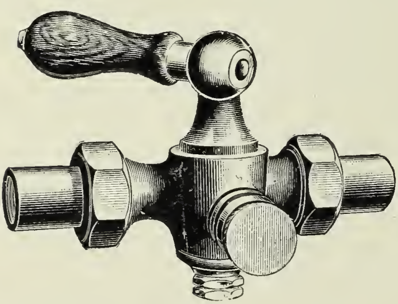


Fig. 10.

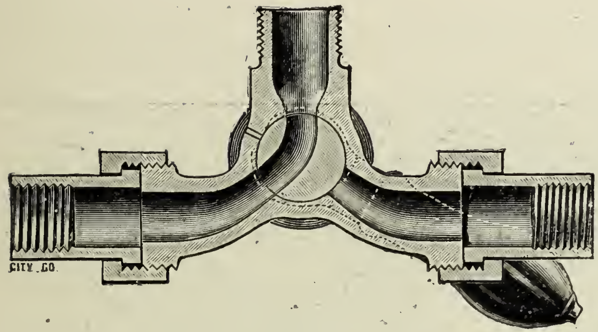


Fig. 11.

Fig. 11 is a sectional view of the three-way cock represented in Fig. 10.

A very simple and efficient cord attachment is shown in Fig. 12, providing a convenient and easy means of adjusting the length of the actuating cord between the reducing gear and the indicator, the hook that is fastened on the cord from indicator connecting in the hole at the end of the attachment.

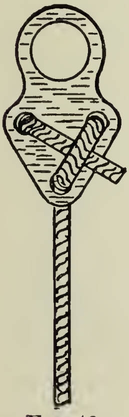


Fig. 12.

Up to a comparatively recent date all indicators have been designed so that the piston spring comes more or less in contact with steam and hot vapors. This increases the temperature of the spring and tends to expand same thereby tending to increase the height of the diagram more than warranted by the steam pressure alone. To overcome this tendency the springs for the ordinary or inside spring type of indicator are graduated and calibrated to give correct pencil movements when subjected to these higher temperatures. The increments of temperature are variable under different working conditions, and, while the inside spring type of indicator as shown in Figs. 5 and 6 will give reasonably accurate results with properly calibrated springs, it is obvious that a better construction would be to so locate the spring as to have it practically free from all such disturbing influences. The Tabor indicator is now made of both the inside spring type as shown in Figs. 5 and 6 and of the outside

spring type as shown in Fig. 13. An examination of this cut, together with Fig. 13A will show that the spring is mounted

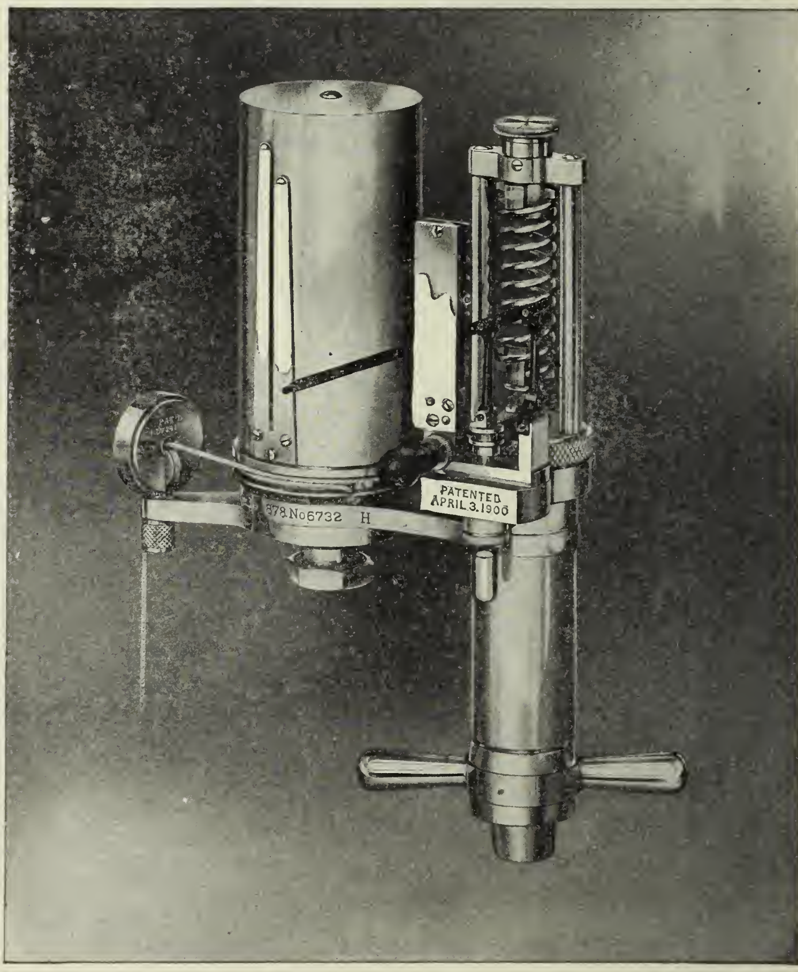


Fig. 13.

above the top cylinder cap where the temperature is but very little higher than the surrounding air. This construction is

desirable for high pressure steam work and especially for gas engine work where the temperatures run very high tending to make the spring unreliable and to suffer deterioration in the inside type of indicator. It will also be noted that with this construction the spring can be much more easily changed and without handling any very hot parts. It is not necessary to disturb the pencil motion or allow the indicator to cool off in order to change springs.

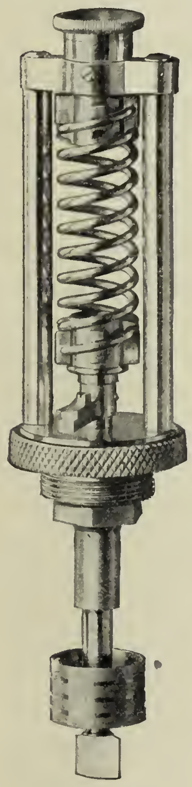
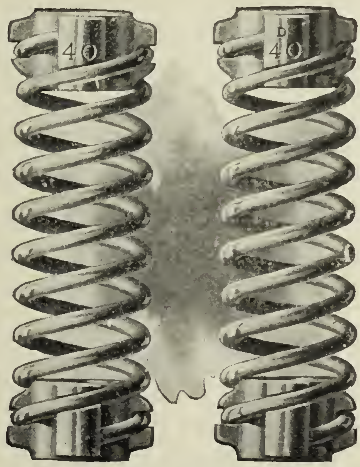


Fig. 13A.

It will be noted from the above that springs for outside and inside types of indicator vary. The general construction of the spring is practically the same for both designs of indicator. See Fig. 13B. Each spring consists of two distinct coils of wire balancing each other, and both mounted with brass screwed ends.



Spring for Inside Spring Style Indicator.

Spring for Outside Spring Style Indicator.

Fig. 13B.

The principal requisites for a reliable and correct spring are, that the wire should be of an exact size, and coiled on a special arbor of such a size that will give the proper tension to the spring for the different denominations. They should be evenly hardened and temper drawn uniformly all over, and in finishing should be made perfectly straight and true, thereby reducing to a minimum all the chances of any

side strain or buckling of the spring, which always tends to force the indicator piston against one side of the cylinder, thereby creating a friction on the piston that will surely cause all results from it to be unreliable. The springs of the Tabor Indicator are made of different lengths, according to the denomination of the spring, the low pressure or light ones being the shorter, and from these continue to increase in length until the highest pressures have been reached. The principal object in making the springs of different lengths is that each may cause the pencil to mark the atmospheric line on the paper drum in an exact position in reference to itself, so as to have an ample range of movement of the pencil either above or below the line for both pressure and vacuum. This variation in the length of the spring obviates the annoyance and necessity of adjusting the length of the pencil connection to locate the position of the atmospheric line in cases where all springs are of the same length, as in some makes of indicators.

CHAPTER V.

INDICATOR APPLIANCES.

It has been found advisable, in order to obtain the best results in indicator practice, to so construct the instrument that the piston shall have only a small amount of motion, and that the movement of the pencil shall bear a certain exact ratio to the movement of the piston.

This ratio varies in different indicators; in some, the pencil has four times, while others have five or six times the piston movement.

This difference in ratio, as a general thing being a matter of selection, or convenience, of the makers in the designing of their indicators.

One of the principal reasons for having the piston move but a short distance as compared with the pencil movement, is that a greater part of the *friction* between the piston and cylinder, due to a long and rapid movement, is *eliminated;* and consequently the results obtained from a short movement of the piston are much more accurate.

By this it is not to be inferred that the shorter the piston movement the greater the accuracy; as there are circumstances connected with the matter that prevent the realization of any theoretical conclusions, in reference to it; because the ratio between the piston might be made so great that the slightest loss motion in the piston connections would be so multiplied at the pencil as to vitiate all efforts to obtain correct results.

On the other hand, by making the ratio too small would tend toward the original principle; where the piston and pencil had equal movement; therefore, the best compromise between the two extremes becomes desirable, and an experience with different proportions seems to indicate that a pencil movement of five times that of the piston, is best adapted to average the imperfections that become involved in either a greater or less ratio.

An important requisite, however, in this respect is, that whatever this ratio may be, is that the pencil shall move exactly that many times the distance moved by the piston throughout its entire range. This result has been perfectly accomplished in the Tabor Indicator through its specially designed pencil mechanism, which causes the pencil to move at any and every part of its travel of exactly five times the distance moved by the piston, and this in connection with the straight line movement of the pencil, as well as the lightness of the moving parts and absence of friction always insures the accuracy of the diagram.

Preparatory to taking indicator diagrams, it is well to see that the piping that connects the engine cylinder with the indicator has been properly done, and in the most convenient position to enable the operations to be performed successfully and with the least amount of anxiety and trouble to the operator. The location of the indicator will, of course, depend somewhat upon the construction of the engine to which it may be applied, but the principal of its operation will always be the same in whatever position it may be placed.

In the indicator piping on the side of horizontal engine cylinders, care should be taken that the holes for the pipe are drilled outside of the point reached by the extreme travel of the engine piston at each end of the cylinder. This precaution is taken to insure that the piston does not close, or even partly close, communication between the inside of the steam cylinder

and indicator piston; which would result in showing, in many cases, a late initial pressure on the card, and otherwise cause it, to a certain extent, to be erroneous. It is also important, in all styles of engines, that these holes should be located as far from the steam ports of the engine as convenient, as there are instances in which a close proximity to said ports has to some extent influenced the pencil from indicating correctly, from the beginning of the stroke to the point of cut off; this being due to the rapid inflowing of the steam through the ports and past the end of the pipe that communicates with the piston of the indicator; thereby causing it to indicate a lower pressure than actually exists in the steam cylinder.

The pipes, preferably of brass, should be as direct as possible and without any unnecessary bends, and in cases where the regular straight indicator cock, shown in Fig. 14, is used at the centre, it is advisable to use straight way valves at the ends of the cylinder.

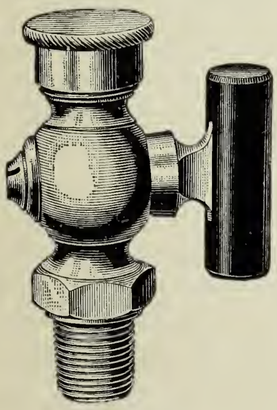


Fig. 14.

There should be no appreciable difference in the mean effective pressure, shown by the cards (under the same conditions), whether the indicator is located at the end or centre of the cylinder, that would be due to any difference in the length of pipe, within the limit of the length of the cylinder. A close comparison of the card from the centre would only reveal a slight increase in the water consumption per horse-power over one taken at the end, and which is entirely due to the added clearance to the cylinder which would ensue from the greater length of pipe in use when the indicator is placed at the centre.

The next matter of importance is in the selection of the best means of giving the paper drum an exact motion coincident

with the motion of the engine piston and cross head, on a sufficiently reduced scale, that will come within the limits of its motion; this distance being limited by the stop formed on the disc (that carries the drum) for that purpose. This entire distance is never utilized in practice, as there must always be a certain amount of allowance at each end of its motion, to guard against any accident that might occur by coming in contact with the stop.

There are various devices for the purpose of giving motion to the paper drum; an example of the most usual ones being shown here in the illustrations, some of which are theoretically correct, while others are only an approximation. The length of the card to be taken, as well as its height, depends somewhat upon the speed or number of revolutions the engine may be running per minute.

In the slow speeds a long and high card may be taken, whereas in the higher speeds a short and low diagram is necessary, in order to avoid as much as possible the effects of *inertia* of the paper drum, and also of the pencil movement. With the new style Tabor Indicator, diagrams can be taken five inches long and two and a half inches in height if desirable, but with the ordinary speeds up to 100, and from that to 200 revolutions per minute, a length of card of 4 inches in the first instance, and 3¾ in the latter, will show well proportioned diagrams. As a guide in this matter it is recommended that with speeds up to 300 revolutions per minute the length of the card may be 3½ inches; of 400, 3 inches; of 500, 2½ inches; and of 600, about 2 inches long, which will insure reliable results. The piston spring should also be of a higher tension (that is, a stronger spring), because, as the speed is increased, it becomes advisable to decrease the height of the diagram in about the same proportion as the length is shortened.

Diagrams taken in these proportions seldom require any change of tension of the drum spring between the highest and

lowest speeds. The lazy tongs shown at about $\frac{1}{6}$ actual size in Fig. 15 is one of the appliances frequently used to obtain the necessary coincident motion (on a reduced scale) between the paper drum and cross head, and which it accomplishes in an accurate and satisfactory manner, provided the device is well made and free from all lost motion in its many pivoted connections. It is usually pivoted at the end (B) by a stud and winged thumb-nut to a block of wood, or an angle iron, secured to the floor of the engine room or in some other convenient position, while the end (A) is fitted in a suitable piece secured to the cross-head of the engine.

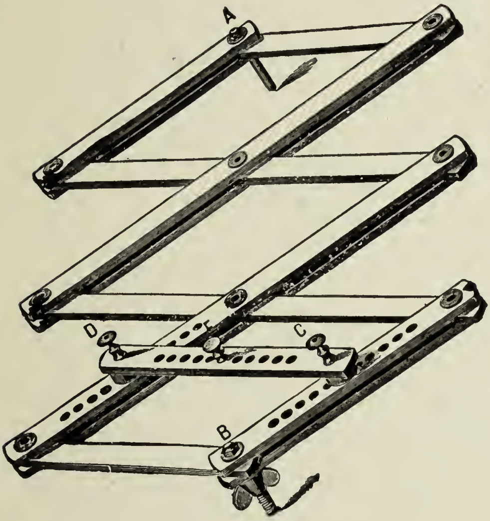


FIG. 15.

The actuating cord from the indicator is attached to the cord pin (E) on the cross bar (C D); said cross bar may be moved in different positions with relation to the centre (B), by changing the screws C and D, which hold it in place, but the cord pin (E) must *always* be placed in a line with the centres (A) and (B).

The position of the cross bar C D, in relation to B, determines the length of travel of the cord pin E, and consequently the length of the diagram. No special care is necessary to locate the position of B, in reference to the cross-head of the engine, so long as the device works perfectly free throughout its range. One of the principal advantages of the Lazy Tongs over some of the other forms lies in its adaptability to the various different conditions so often found in indicator practice.

Fig. 16 represents one way of attaching it to an engine, when the indicators are placed on the side of the cylinder. In this case it is worked in a horizontal (or flat) position, one end is supported by a standard secured to the floor, and of sufficient

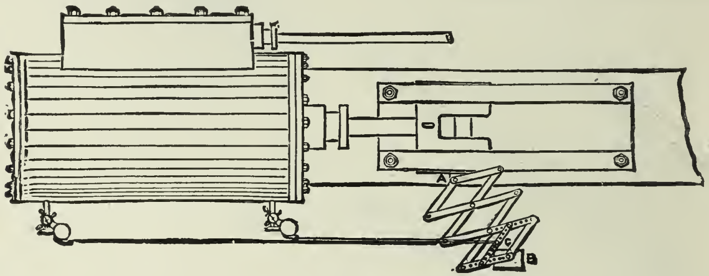


Fig. 16.

height to just bring the cord pin (E) on a level with the cord guide on the indicator; consequently, in this case, the cord will run *direct* from the cord pin to the *indicator*.

They may also be used in a vertical position with equally good results, where the end B is attached to a low block or bracket secured to the floor, and directly below the engine cross-head. In this position it becomes necessary (in order to insure coincidence between the cross-head and paper drum) to use a small carrying pulley, over which the cord *must* pass from the cord pin, E, and thence to the indicator. Said pulley may be mounted on an additional suitable block that will admit of its being placed exactly on a level with, and a short distance from the cord pin, E.

There are various other ways of applying the Lazy Tongs, as circumstances may require, which will suggest themselves to the engineer, all depending entirely upon occurring conditions.

Proportion of Lazy Tongs. In order to enable the engineer wishing to construct an accurate device similar to that shown in the illustration, the following data will need to be carefully observed to insure an accurate motion.

The instrument is constructed principally of strips of thoroughly seasoned cherry wood about one and one-eighth (1⅛) inches wide, by three-eighths (⅜) inch in thickness.

The distance apart of the outer holes in the long strips is sixteen (16) inches, while the length of the short strips connecting their ends are only one-half of that, or eight (8) inches between similar holes.

In the cross bar C. D. there are eleven additional holes, one-half (½) of an inch apart, placed equidistant from each end, and are threaded to admit of the cord pin E, being screwed therein in its various positions on the cross bar.

In one of each of the long and short strips are also eleven threaded holes, one-half (½) inch apart and exact duplicates of those in the cross bar C. D. and to which the cross bar is secured by screws in any desired position, as shown in the illustration.

The pivoted joints are constructed of brass tubing about five-sixteenths (5-16) inch outside diameter, by three-sixteenths (3-16) inch inside, and of sufficient length to go through the joints and be riveted over iron washers on each end.

This construction also provides a means of tightening the pivots in case they become loose from wear, by a further riveting over of the tubing.

There is also placed upon the pivot between each joint, two thin brass separating washers to prevent the strips of wood coming in contact with each other while in operation; thereby obviating the friction that would otherwise take place.

The studs A and B are shown one-half size in Fig. 17 and are made of round steel, seven-sixteenths (7-16) of an inch in diameter, and three and one-half (3½) inches long. In the middle of their length is secured a collar three-quarters inch in diameter and one-quarter of an inch in thickness. Each of these studs on one side of the collar are made with nut and washer, and serve as pivots to connect the ends of the device.

The opposite end of stud A is made either straight or tapering as occasion requires, to connect with the cross-head; while the opposite end of stud B, is threaded, and provided with winged nut, for the purpose of attaching to the fulcrum, or to whatever the device may be suspended; as before mentioned.

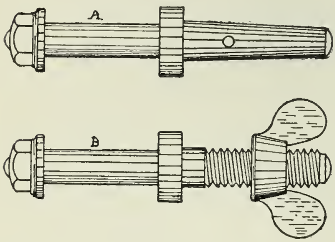


Fig. 17.

In order that the cross bar may not come in contact with the nut and washer of the stud B, when the device is closed, the bar is elevated by means of two wooden washers, one at each end, made of the same material as the strips, and the screws for securing the same are of sufficient length to reach through both, and screw into the drilled strips underneath.

It is exceedingly important that all corresponding holes in the strips be layed out and drilled with extreme accuracy, and also the pivots made a close fit therein, to insure its perfect working at all points, from its closure to the full extension.

In this construction of the Lazy Tongs, the location of the holes, their distance between centres, and the position of the cross bar in relation to the stud B, determines the ratio (that occurs in each case) in the amount of motion of the cord pin E, as compared with the amount of motion of the stud A.

The holes being one-half inch apart, a change of the cross bar from one hole to the next, results in a change of motion of the cord pin E, of one-forty-eighth.

For instance; If the cross bar be secured in the holes that will bring it nearest to the stud B, and if the cord pin E, there be placed in the hole that comes in line between the studs A and B, the reduction of motion of the cord pin will be three-forty-eighths, or one-sixteenth of that of the stud A; in the second hole the reduction is four-forty-eighths, or one-twelfth; in the third hole, it is five-forty-eighths; in the fourth hole

it is six-forty-eighths, or one-eighth and so on, to the farthest holes from B, where the motion of the cord pin becomes thirteen-forty-eighths of that of the stud A.

From this it will be seen that each position of the cross bar has its own ratio of movement between the cord pin, and stud A, as above, whatever the length of stroke of the engine may be.

In every position of the cross bar C. D. (if made according to directions), there is *always one of its holes* on an exact line between the centres of the studs A and B, and into which the cord-pin E, must always be placed in order that it may move in an exact straight line parallel to the motion of the stud A.

CHAPTER VI.

INDICATOR APPLIANCES CONTINUED.

An accurate and simple form of pantagraph is shown in Fig. 18, in which the end, A, is connected with the engine cross-head by means of a pin or other suitable connection; while the end, B, is pivoted to a bracket, C; said bracket also serving as a support for the guide pulley, E, and upon which it may be adjusted at a level to coincide with the cord pin, D.

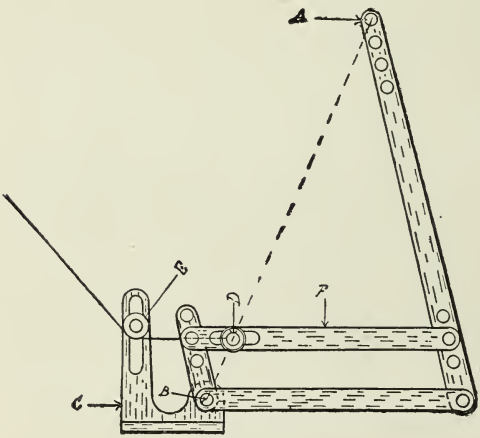


Fig. 18.

The length of travel of the cord pin depends entirely upon the distance the cross-bar, F, may be placed in relation to the fulcrum, B. There is a slot in this cross bar which admits of an adjustment, and the securing of the cord pin, D, in any required position within the length of the slot; and in whatever position the cross bar may be placed, the cord pin must *always* be moved and secured by means of the thumb nut, on a line between the points A and B in order to insure an exact straight movement of the pin, D, and parallel with the engine cross-head.

This instrument, like the Lazy Tongs, may be used in either a horizontal or vertical position; it has a less number of pivoted connections and accomplishes equal accuracy; but has not quite the same range of application under all circumstances. In all cases and with either instrument, where they are placed in a position requiring the use of a carrying pulley, care must be taken that said pulley, E, be located a short distance from the instrument, and in such position that the actuating cord will move parallel with A, from the cord pin, D, to the pulley, E, and thence in any desired direction to the Indicator.

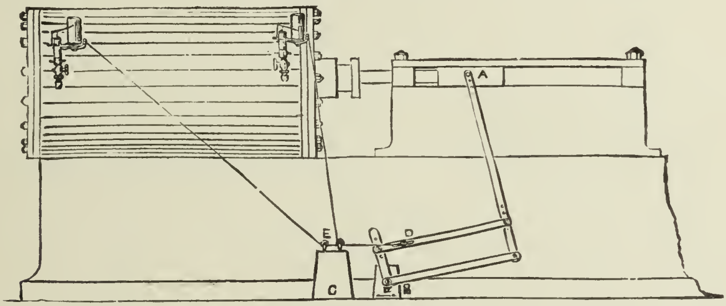


Fig. 19.

Fig. 19 represents the pantagraph in its application to an engine cylinder, where the indicators are placed upon the side of the cylinder. In this case blocks of wood or iron is shown, secured to the floor, to serve as supports for the instrument, as well as the carrying pulleys; this arrangement answers the purpose, but the bracket, C, in Fig. 18, is much more mechanical.

A simple method of obtaining the drum motion, and very frequently used by engineers (in the absence of a more accurate device), is the pendulous lever, A B, shown in Fig. 20, which consists of a strip of wood, about 4 inches wide, by ⅞ inches thick, and suspended from a bracket, G, which is secured to the ceiling or any overhead framing; the lever being of sufficient length from its point of suspension, so that its lower end,

A, may connect to one end of the connecting bar, C, while the opposite end of said bar is attached to the engine cross-head, D. The length of the lever, A B, should be such that the end, A, should be as far *below* the line of movement of the cross-head, when on its center of travel, as it will be *above* the line at its extreme end of stroke.

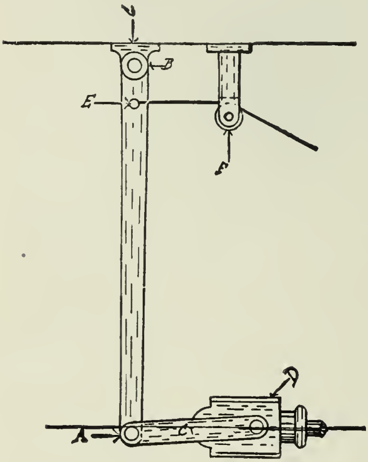


Fig. 20.

The cord pin, E, (to which one end of the indicator cord is attached), is located at such a distance from the point of suspension, B, as will rotate the paper drum, an amount that will give the desired length of the diagram. In this device it is necessary to pass the cord from the pin, E, over the guide pulley, F, and thence to the indicator. In connecting the device, it is important that the cross-head, D, should be at its center of travel, when the lever, A B, is in a vertical position.

This reducing motion may be easily and cheaply constructed and set up by any engineer, and will be found a convenient means of giving motion to the paper drum; but although *not mathematically correct*, will give fair results.

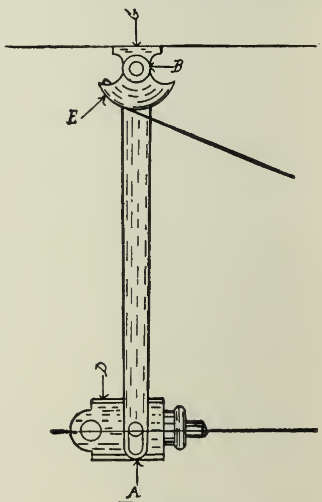


Fig. 21.

Fig. 21 shows another device, the same in principle, but instead of the cord pin there is substituted a segment of a circle, having its center at B, and with a radius necessary to give the paper drum the desired amount of motion.

In place of the connection, C, as in Fig. 20, the lower end of the lever at A is slotted and is worked by a stud, secured to the engine cross-head, D. Whenever this device can be placed in a direct line with the indicator, the guide pulley can be dispensed with, and the cord encircling the segment may be run from it directly to the indicator.

As the relative length of the lever in this arrangement is constantly changing throughout its stroke, it is therefore as to the matter of inaccuracy about equal to Fig. 20.

The plan of suspended lever, shown in Fig. 22, is an improvement on the previous ones shown, and was proposed by Mr. Frank Richards in an article published in the *American Machinist* several years ago, in which he gives a number of modifications of the same principle, whereby perfect theoretical accuracy is attained. It consists of an ordinary suspended lever, with the lower end, A, slotted and driven by a pin secured to the engine cross-head; the end, B, being pivoted to the bracket,

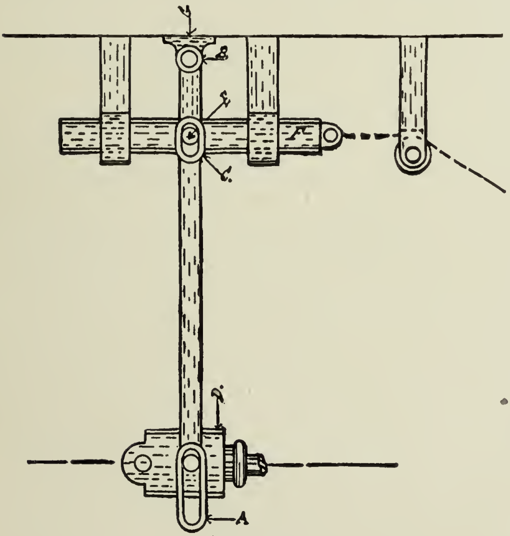


Fig. 22.

G, which may be secured to the ceiling or some suitable frame work (or it may be used in a horizontal position if desired); the other element of the lever at C also being slotted, and giving motion to a sliding bar, F, by means of a pin fastened to said bar, and working freely in the slot in the lever. The bar, F, slides in hangers secured in the same manner as the bracket, G, and in a line parallel to the line of the cross-head. The

cord, attached to the sliding bar, passing in a line over a suitable guide pulley and thence to the indicator. By this arrangement the relative length of the elements of the lever, A B, always remains the same, and consequently insures coincidence and accuracy of the drum movement, throughout its entire range.

The various different modifications of this principle, to attain accuracy of the drum motion, depends upon the use of the sliding bar, F, in each instance, and it is evident, as well as readily seen, that it makes no difference (so far as accuracy is concerned), whether the lever, A B, is slotted, or instead of the slots, we substitute two permanent pins in the lever, and have each pin work in slots formed in both sliding bar, F, and in a suitable slotted piece secured to the engine cross-head.

Figs. 20, 21, and 22 show the position of the lever, A B, when engine cross-head is at the center of its stroke. In order to determine the proper distance (of the point on the lever to which the cord is attached), from the fulcrum, B, to take any desired length of card, it is necessary first to divide the stroke of the engine cross-head, in inches, by the desired length of the card required; then divide the total length from center to center of the lever, A B, by that quotient, which will give the distance of the point in inches from the fulcrum, B, to the pin, E, or where the cord must be attached. For example, suppose the stroke of the cross-head to be 24 inches, the length of the lever, A B, between centres to be 48 inches; and we desire to produce an indicator card 4 inches long; then 24 inches divided by 4 inches, the length of the desired card, is equal to 6, and by dividing the length of the lever, A B, which is 48 inches by 6, gives us 8 inches, which is the proper distance from the fulcrum, B, to a point on the lever to which the cord must be attached to produce a card 4 inches in length.

A slight discrepancy will sometimes appear in the length of cards, computed from any rules, owing to incorrect measurements, stretching of the cord, inertia of the paper drum, &c.

Fig. 23 shows another modification of the lever and sliding bar, F, in which the slots in the lever have been dispensed with, the lever being attached to the cross-head and bar F by means of the connections A H and C E. An important matter in this construction is that the lengths of the two connecting links must bear the same ratio to each other, as the ratio between the two elements of the lever, A B and B C; therefore A B : B C :: A H : C E. The pins on the cross-head and the sliding bar must be so located that lines drawn through the centres of the connections A H and C E, will be parallel with each other in any position the lever may assume.

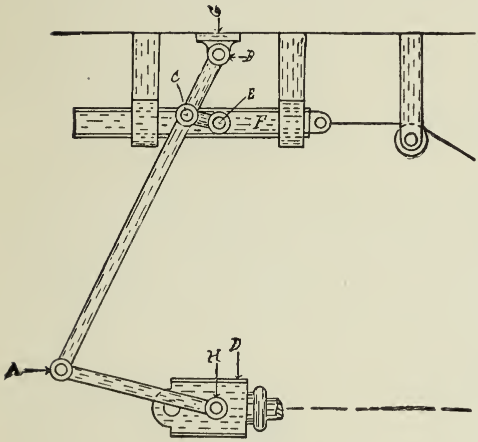


Fig. 23.

In the arrangement shown in Fig. 24 the fulcrum of the lever is at the pin, C, of the sliding bar, F, while the ends are each made in the form of a fork, the end, A, being moved by and sliding upon a pin secured to the cross-head, the end, B, moving on a pin fastened to the bracket, G. In Figs. 23 and 24 the cross-head is represented at the extreme end of the stroke.

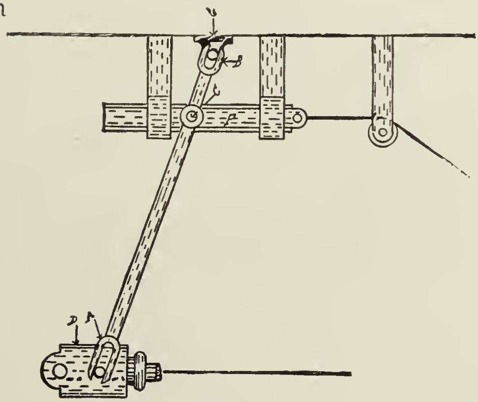


Fig. 24.

These are modifications of Fig. 22, but without claiming any particular advantage for one over the other. In all cases where this principle is accurately applied, it will be found that the elements of the lever, A B, are always in a line at a right angle to the line of movement of the engine cross-head, at the time the cross-head is at the centre of its travel.

The sliding bar principle may also be well adapted to engines where the end of the engine shaft is accessible. By inserting a stud, A, in the end of the shaft, as shown in Fig. 25, at a proper distance from the centre, C, to give the desired length of card, and connecting it to a sliding bar, F, by means of a short connection, A B, will insure a perfect coincident motion of the engine cross-head on a reduced scale. The bar may be extended to any desired length to secure convenience in connecting with the indicator. A necessary requisite, to insure correctness by this arrangement, is that the length of the connection, A B, relative to the distance of the end, A, from the centre, C, must bear the same proportion that the main crank of the engine bears to its own connecting rod. It is not necessary in this case that the slide move parallel with the movement of the cross-head, as it may be moved in any desired direction, but great care must be taken that stud A be placed in such position on the shaft that the sliding bar, F, will be on its extreme throw at the same exact time that the engine crank is at its extreme movement, and both at the same end of the stroke. Where the end of the shaft is not accessable, the same results may be accomplished by means of an eccentric placed in a convenient position on the shaft.

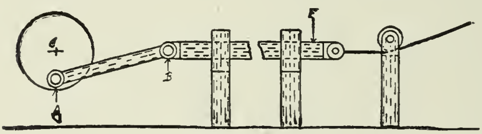


FIG. 25.

A reducing arrangement, that gives good results at high speeds, is a plain lever swinging on a pin above (or it may be below) the cross-head, having a pin in the end which slides up

and down in a slotted piece of metal fastened to the cross-head as shown in Fig. 26. This method gives a fairly accurate motion. A segment of a grooved pulley is fastened to the

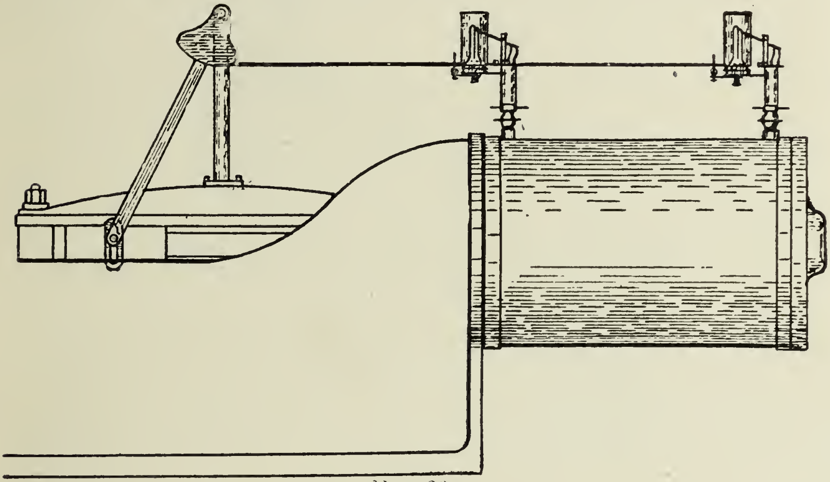


FIG. 26.

upper end of the lever, and from this the cord extends to the indicator.

The whole device should be made of wood as light as possible, (consistent with strength), in order to reduce the inertia to a minimum. A heavy reducing gear on a high speed engine will wear very quickly and create inaccuracies in the diagram.

When indicating High Speed Engines or Locomotives, the driving cord for hooking on the indicator should be continued beyond the loop, and fastened to a spring or an elastic band attached to the carrier pulley of the indicator.

This band or spring is intended to always keep a tension on the driving cord, whether the indicator is in operation or not, and prevents entanglement and breakage of cord when the indicator is unhooked.

The cut in Fig. 27 shows the arrangement as described, and which operates in a perfect satisfactory manner, as diagrams can be taken with as little difficulty at high speeds, as on slow speed engines.

The hook, A, on the indicator cord connects into the loop, B, on the driving cord when the indicator drum is in operation;

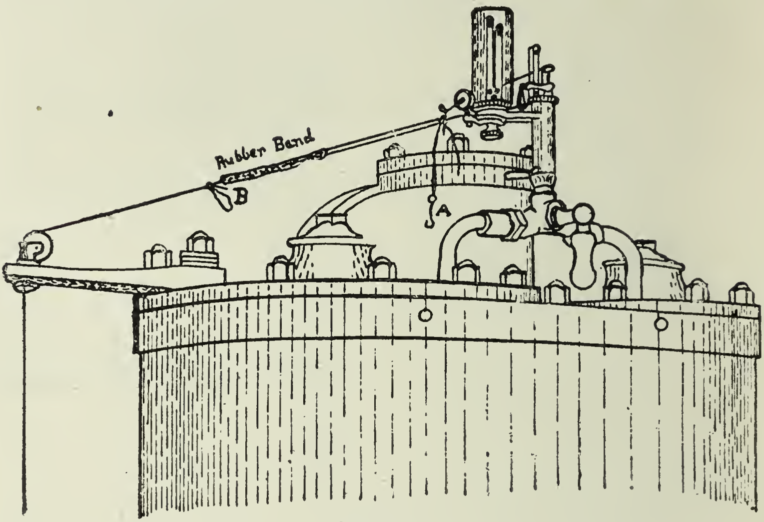


Fig. 27.

the loop, B, is made long enough to be held with the hand when hooking on the indicator.

To disconnect, merely catch the hook and hold it stationary for a second, and the loop will come off. The rubber band then takes the motion and keeps the cord taut.

Whatever arrangement is employed, it is desirable to avoid the use of long stretches of cord on account of its sagging and stretching. Small wire may be used to good advantage on vertical lengths except where the line passes over pulleys.

Whatever the style of reducing motion that may be employed for giving motion to the indicator drum, its accuracy can easily be tested, and ascertained in the following manner· Lay off a number of *points* on the cross-head guides, at say ⅛, ¼, ⅜, ½, ⅝ in. etc., of the stroke.

Connect the Indicator with the reducing motion in the same manner as for taking diagrams. When the cross-head

is on either dead centre, bring the pencil in contact with the paper on the drum and make a short vertical line.

In the same manner make other lines on the paper, as the cross-head is moved to each successive eighth point on the guide.

Then if the lines on the paper are exactly at eighths, the motion of the cross-head has been accurately reduced.

These directions given for reducing motions are general; some special cases require special modifications.

CHAPTER VII.

INDICATOR REDUCING MOTION.

There are a number of devices under different names, all designed for the purpose of giving the necessary accurate motion to the paper drum; and each constructed on principles in which long swinging levers are dispensed with, the actuating cord from these devices being connected either directly to the engine cross-head, or to some moving part that has a motion in exact unison with it; while another and separate cord connects the device with the indicator. If, in the act of operating the indicator with any of these devices, it becomes desirous of stopping the motion of the paper drum, it is necessary to disengage, or unhook, either one or the other of the cords that give it motion; a matter with some of them, requiring considerable

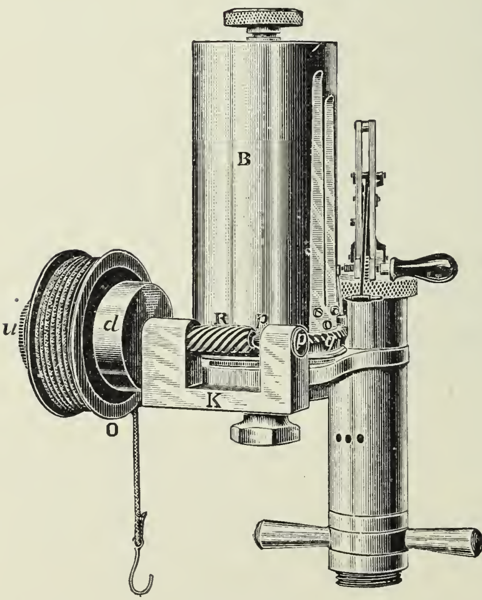


Fig. 28.

practice to accomplish with ease and assurance especially on engine having high rotative speed. This operation of unhooking is usually performed on the cord connecting the device with the Indicator. Some of these attachments are constructed so as to be attached to the indicator by means of a bracket, adapted to the particular indicator to which it is to be applied; while others are so made, that may be secured away from the indicator, to some part of the cylinder or engine framing by means of set screws or bolts; in many cases requiring considerable labor to secure them in a convenient position.

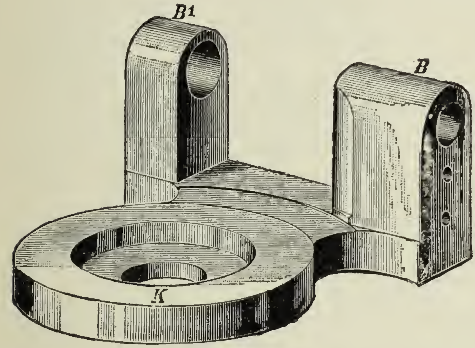


Fig. 29.

The particular Reducing Gear, of which we shall give a description in this article, is one that is constructed with and forms a complete part of the well known Tabor Indicator, as illustrated in Fig. 28. After securing the indicator in position on the pipe conneeting with the engine cylinder, and attaching the end of the actuating cord to a stud screwed in the cross-head, the instrument may be used during the time of any experiment, without the necessity of connecting and disconnecting the actuating cord, in order to start or stop the motion of the per drum.

The principal parts of this Reducing Gear consists of supporting base K, Fig. 29, with two short standards, B, and B'. The standard B' is fitted with a hardened steel center P, Fig. 30, which serves as a pivot bearing for the end of the worm shaft R, Fig. 31, and which receives the entire thrust of the shaft R, thus reducing the friction from that factor to a minimum; the other bearing being in the standard B. The base

K is connected direct to the Indicator, upon the projecting arm that supports the paper drum B, and the teeth of the worm shaft, R, engage directly with the teeth on the drum carrier g, thereby making a positive connection therewith, and forming a part of the Indicator, (see Fig. 28). To the base K, is also connected the spring case D, Fig. 32, permanently secured thereto, by means of screws passing through holes as shown in the spring case, and corresponding with holes tapped into the standard B, (as shown) of base K, and so located thereon that its centre shall be common with the center of the worm shaft R.

Fig. 30.

Fig. 31.

Upon the worm shaft R, is secured by means of a set screw, the collar A, through which freely slides the clutch pin I, one end of which is securely fastened to the thumb piece U, by which the pin is operated. The whole mechanism of that part is shown complete in Fig. 33. The flanged pulley O, Fig. 34, rotates freely and independently forward and back on the worm shaft R. It has its outer hub formed in the shape of a double cam clutch, with steel pins X, X, inserted to prevent wear upon their faces, while the opposite side has a hole in which the pin A, of the spring case cover S, Fig. 35 engages. One end of the actuating cord is attached to the pulley O, (a hole being shown in the figure for that purpose) while the other is secured either to the engine cross-head; a

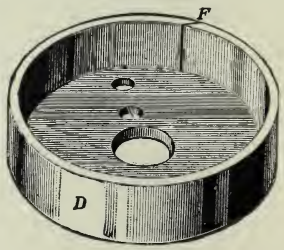


Fig. 32.

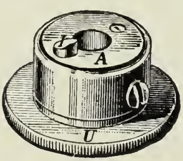


Fig. 33.

standard bolted to the same, or to any other part that has an exact similar movement, and should be connected from the pulley O, in a line parallel to the movement of the cross-head.

The length of the actuating cord should be such, that when connected to the cross-head, at the *commencement of the outward stroke* of the engine, there should also be about *six or seven turns* of the cord encircling the pulley O, which will always insure the requisite amount of cord, and obviate the liability of breaking.

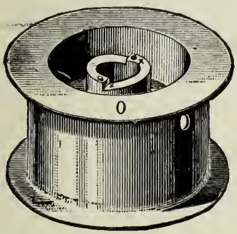


Fig. 34.

Enclosed in the spring case D, Fig. 32, (not shown in the illustration) is a small plain Spiral Spring; one end of which is secured to the spring case by means of a slot F, shown in the figure, while the other end connects with the hook C of the spring case cover S, Fig 35. This spring operates to return the pulley O, back to its starting point, after it has been revolved in one direction, by the outward movement of the engine cross-head; therefore as this pulley O, has an independent rotating forward and back motion on the worm shaft, R, the necessity of unhooking the cord in order to stop the motion of the paper drum B, after the diagram has been taken, will be entirely overcome, as explained further on.

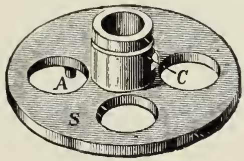


Fig. 35.

The paper drum B, is rotated *forward* by means of the pulley O, through the worm shaft R, engaging with the teeth of the gear g, on the drum carrier; and in the *opposite* direction by the action of its own retracting spring. On the top of the drum B, is a knurled thumb-piece, with a projecting pin on its under side, for the purpose of engaging with a similar pin, secured in the top of the drum, and is to be used by the operator when in the act of starting the drum; for the purpose of moving it slightly forward before the clutch pin I, is pushed in engagement with

the cam hub of the pulley O, thereby preventing the drum carrier from striking against its stop on the return motion.

With this reducing gear the stopping of the drum motion becomes a very simple matter, and is accomplished by taking hold of the thumb piece U, and withdrawing the engagement between the clutch pin I and the clutch hub of the pulley O.

The knurled thumb piece on the top of drum B, also furnishes another very convenient (and preferable) means of stopping the motion of the drum; as by holding it so as to retard the motion of the drum on its return stroke, it will thereby cause the cam face of the pulley O, (which is constantly in motion) to automatically force the clutch pin I out of contact with the clutch hub of the pulley O. The starting or stopping of the paper drum, *at any time*, will have *no effect* on the motion of the pulley O, which will continue to revolve independently while the engine is in motion. When desirous of starting the drum B, it will be necessary to *again* make the engagement between the clutch pin I and the clutch hub of the pulley O, and which must be done by the combined means of the thumb piece U, and the thumb piece on top of drum B, as follows: The pulley O, being in constant motion; with one hand take hold of the thumb piece on top of drum B, and turn it in a direction from right to left, until it carries the drum and its carrier a short distance from its stop, (say about ¼ inch). While holding the drum in this position, take hold of the thumb piece U, with the other hand and gently press it toward the clutch hub of the pulley O, and it will be found that when the engine cross-head arrives at its extreme inner stroke, that the engagement between the clutch pin I and the clutch hub of pulley O will *always* take place at that particular point, and with the least amount of difficulty in the operation.

The thumb pieces are so constructed that they may be readily held in the hand while running; therefore no difficulty is experienced in throwing the clutch pin in or out of gear. In

preparing to use this Reducing Gear, the first and very important matter for consideration is in the selection of the pulley O, which shall be of such size in relation to the stroke of the engine as will give the requisite length of the Indicator Diagram.

A ready and mental means of ascertaining this, is by dividing the length of the stroke of engine (in inches), by twelve, which will give the diameter (in inches), of the pulley O, that is, the stroke of the engine being 12 inches will require the pulley O to be one inch in diameter; a stroke of 18 inches, 1½ inch; of 24 inches, 2 inch; and of 36 inches a pulley of 3 inches in diameter, &c.; and it is also well in this connection to bear in mind that for *each complete revolution* of the worm shaft R, there will a corresponding pencil line (when in contact), marked on the paper drum horizontally, about *one inch* in length; consequently, two revolutions mark two inches; three revolutions gives three inches, length of card, &c.

It must be understood that this calculation of the size of pulley O, has reference to engines having low rotative speed, say up to 120 revolution per minute, and where a long card is desired; but when the speed is increased from this to 500 or 600 revolutions per minute, the size of the pulley O will have to be in accordance with the recommendation (in chapter V) in reference to the *height* and *length* of diagram advisable, where the rotative speeds are gradually increased up to 600 revolutions per minute, and producing a card two inches in length.

After selecting the pulley required, *remove* the clutch from the worm shaft R, by slacking the set screw shown in collar A, and place the pulley thereon, taking care that the pin-hole on the inner side of the pulley (not shown in cut), engages with the projecting pin A, of the spring case cover S, Fig. 35, then replace the clutch on the shaft as far as it will go and secure it firmly in place by the set screw in collar A.

Place the indicator in position, and attach the actuating cord as already described.

The *tension* of the spring in the spring case D, must be sufficient at all times to just keep the cord taut, and this may be regulated by taking more or less extra turns of the cord around the pulley O, until it results in the desired tension. The alignment of the Indicator, in order to have the cord run evenly, is a matter to be observed; and if upon starting the engine the cord should run unevenly on the pulley O, it may be entirely remedied by slacking the indicator connection a trifle, and turning the Indicator slightly in the necessary direction until a perfect and uniform winding of the cord is obtained, and which can (by this means) *always* be accomplished.

The peculiar construction of the angular teeth on the worm shaft R, and also on the drum carrier G, enables each to become the driver of the other; that is, on the outward stroke of the engine, the drum carrier is driven by the worm shaft; while on the return or inward stroke, the worm shaft is driven by the drum carrier g, through the action of its retracting spring; said spring *always* serving to return *both of them* back to their normal positions, at the extreme inner stroke of the cross-head, at each revolution while in operation.

In the absence of a proper understanding of the principles of manipulating some of the various Reducing Motions in the market, some engineers that have had no particular experience with them, look upon *all such devices* as complicated and generally troublesome to operate and are satisfied to continue the use of the more antiquated pantagraph and lever movements.

In our own experience and that of engineers who have been using the positive Reducing Gear, described and illustrated in this article, it may be said to have given entire satisfaction in all cases, and a return to any of the old methods, (after becoming familiar with this) could not be contemplated under any circumstances.

CHAPTER VIII.

DRUM STOP MOTION AND ELECTRICAL APPLIANCE.

In using the different lever and pantagraph devices, that have been illustrated in a previous chapter (or any modifications of them), for the purpose of giving motion to the paper drum; it becomes necessary with any and all of them to connect and disconnect the actuating cord leading from the device to the indicator, in order to start or stop the paper drum whenever necessary, for either adjusting or the removal of the paper from the drum. In slow speed engines, the matter of hooking and unhooking the actuating cord is of no great difficulty;

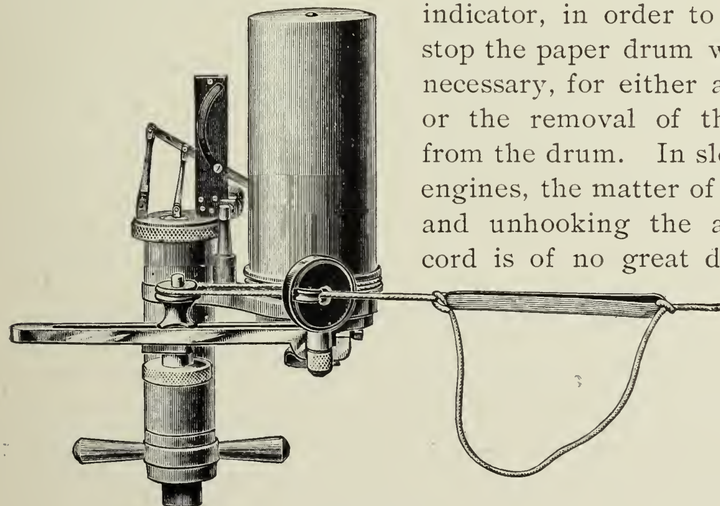


Fig. 36.

but with engines of high rotative speed, it becomes more difficult and requires much more skill and experience on the part of the operator to perform the operation successfully and with

ease. An efficient and very simple attachment, which may be adapted to different styles of indicators, for the purpose of starting and stopping the paper drum at all times without the necessity of unhooking the actuating cord, when used in connection with any of the pantagraph styles of reducing motion, is illustrated in Fig. 36, attached in this case to a Tabor Indicator; and whereby the usual handling of the actuating cord (that otherwise becomes necessary), to stop the motion of the paper drum, is entirely obviated at any and all speeds.

The device is shown in Fig. 37 detached from the indicator, and consists of an arm A, which may be secured to a part X of the indicator, by means of the set screw B. Upon the arm A, Fig. 37, is a slide C, which may be adjusted to any desired position on the arm and secured thereto by the knurled nut E and washer F. On the slide C is mounted the cord pulley D for the purpose of directing the actuating cord (from any form of reducing motion), around the said pulley and thence on to the paper drum carrier to the indicator.

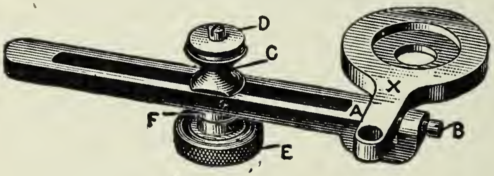


Fig. 37.

The method of determining the proper length of the actuating cord is as follows: set the slide C, to its extreme inner position on the arm A, and place the engine on its extreme outer stroke, then bring the cord from the drum carrier around the cord pulley D, and thence in the required direction to the point of attachment on the reducing motion, which will give the necessary length of the actuating cord. At any convenient position on the actuating cord and near the cord pulley D, there is superposed an elastic band, shown in Fig. 36, for the purpose of taking care of the slack of the cord, that will appear when the engine is in motion and the paper drum at rest. This slack is owing to the length of the actuating cord being

taken when the engine is at its extreme outer stroke; consequently on its arrival at the inner stroke there is an amount of cord to be taken care of, equal to the presumed length of the indicator diagram, and the intervention of the elastic band is for that purpose only.

While the slide C is at its inner position no motion will be transmitted to the paper drum; but by moving the slide C outward upon the arm A, it will at once cause the paper drum to rotate forward and back, and the slide may be secured in any desired position on the arm by means of the knurled nut E, therefore in order to start, or stop the motion of the paper drum at any time when the engine is in motion, it is only necessary to change the position of the slide C on the arm A, which is accomplished most satisfactorily, by means of the knurled nut E.

To start the paper drum, move the slide outward on the arm and secure it by the nut E, and to stop, move it to its extreme inner position, the actuating cord continuing its usual motion during the time the engine is in motion.

To Take Diagrams Simultaneously. In order to make complete and reliable tests of steam power from the various compound and multiple cylinder engines, or whenever it is desirous to take diagrams *simultaneously* from a number of steam cylinders (in which as many indicators are used), it becomes necessary to provide some sure means of operating the indicators, by which an operator can accomplish the object *alone*, without the aid of assistants and with the certainty that *all diagrams* taken at any particular stroke of the engine or engines, will all commence and leave off simultaneously with each other, and thereby dispensing otherwise with the number of attendants necessary to operate the different indicators at some decided upon signal; a plan whereby an exact coincidence of the diagram at any particular stroke is very *rarely obtained* owing to the almost impossible concerted action between the operators.

The taking of diagrams from two or more cylinders at the same exact stroke of the engine or engines, may be accomplished successfully in different ways that will oft-times, as occasion requires, suggest themselves to the engineer. An arrangement sometimes used with fair success, consists of a small reservoir of compressed air from which the pressure is communicated to a small piston within a cylinder secured to each indicator; one end of the *piston rod* being in contact with some part of the pencil mechanism, consequently any movement of the piston is communicated directly to the said mechanism, which results in a contact (when under pressure from the reservoir), between the pencil and paper drum; and a withdrawal of the pencil, (through the action of a spring) when the pressure is released. These small cylinders are connected to the reservoir by means of rubber tubing in which there is a cock for admitting and releasing the air pressure upon the piston. The operation of the device being about as follows: after seeing that the parts are properly connected, start the pencil mechanism of the indicators in motion, by opening the cock connected to each, then by opening the cock from the reservoir the pressure from that source (through the small piston and its rod), will force the pencil in contact with the paper drum and against the resistance of the spring. The time of contact between the pencil and paper drum is supposed to be during one complete revolution of the engine, unless an average card is desired from a number of revolutions. The instantaneous release of the air pressure against the small pistons, takes place in the act of closing the cock; it being provided with an escape hole for that purpose, thereby admitting of the spring to at once move the pencil out of contact with the paper drum.

In the absence of a reservoir of compressed air, the same results may be obtained in this device, by using a jet of steam through a small pipe leading from the boiler, steam pipe, or from any convenient place where a pressure of steam may be

obtained. Other improvised means depending upon circumstances and the ingenuity of the engineer, may be used to produce the desired result. The most successful, simple, and satisfactory results, may however, be obtained by the use of the electric current. A very neat and simple electrical attachment to enable an operator to produce diagrams, from any number of cylinders during the same stroke of the engine by simply pressing a button to close the electrical circuit is represented in Fig. 38, as attached (in connection with the reducing motion), to the well known Tabor Indicator.

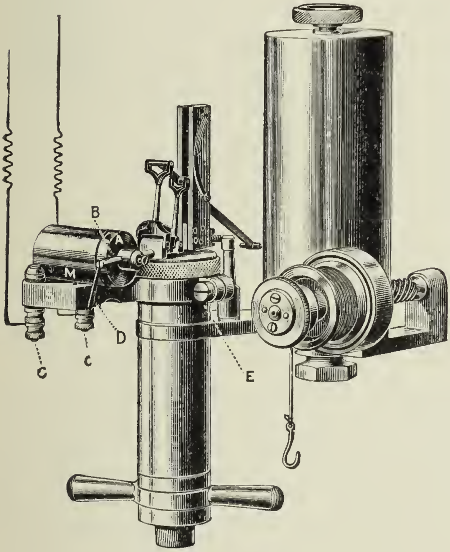


Fig. 38.

The attachment consists of a specially constructed magnet M, mounted on and secured to a support S, which encircles the body of the indicator, and is held in position by the clamping screw E. Also secured to the support are the binding screws C and the spring D, these parts being shown separate from the indicator in Fig. 39.

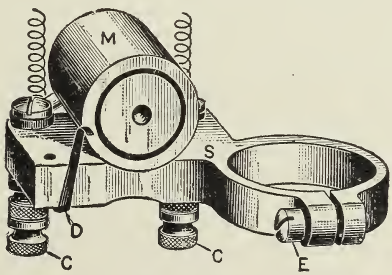


Fig. 39.

The stud B, Fig. 40, is screwed into the upright on the swivel plate that carries the pencil mechanism of the indicator and serves as a support for the armature A, Fig. 41, and to which it is secured by a small set screw for that purpose; therefore any movement of

the armature A, in either direction, relative to the magnet M, produces a similar motion of the pencil (in the opposite direction), to or from the paper drum of the indicator.

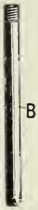


Fig. 40.

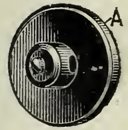


Fig. 41.

The spring D, Fig. 39, is formed so as to hold the armature within the field of the magnet, before the current is established, and also to quickly release it when the current from the battery is broken. The magnet M, consists of a *single spool* of soft iron, wound in the usual manner with insulated magnet wire, and enclosed by a soft iron shell, the combination thereof establishing the two poles of the magnet, when subjected to the effect of an electric current passing through the wire. The armature A is also constructed of a soft grade of iron and is finished to a diameter the same as the magnet shell, and is adjustable on the stud B, to exactly coincide with the magnet M. On the side facing the magnet are two small brass pins for the purpose of assisting in the instantaneous release of the armature, from the magnet after the current is broken. This electrical device is easily attached or detached in a few seconds, and its connection with the indicator, does not in any way interfere with the usual manipulations of the operator, in adjusting the paper to, or removing it from the paper drum; changing of the springs in the instrument, or any minor operations that may become necessary, as the pencil mechanism is free to be revolved in any convenient position.

It is represented in Fig. 38, as being attached to a Right-hand Indicator, but it may be used on a Left-hand instrument with equal facility. The change from one to the other is easily and readily made in the following manner: first unscrew the cap (that carries the pencil mechanism), and take it complete from the indicator, then by loosening the clamping screw E, the support S may be readily removed, and it only remains to unscrew the magnet M, from the support S, and change the

location to directly the opposite side of the support and secure it in that position, by means of the small screws for that purpose. The binding screws, as well as the spring D, will also need reversing on the support S; the parts all being provided with means by which it may (if necessary) be easily and readily accomplished. Replace the device on the indicator in a reversed position and secure it by the clamping screw E. Where the circuit is short the device may be operated in connection with a single indicator, by any *one* of the well known batteries in the market, (either dry or liquid).

In our own experience, and in all cases where such an appliance is desired, and where accuracy is necessary, this simple electrical device seems to meet all requirements, being easily attached or detached, without any change in the mechanism of the indicator; instantaneous in its action, and can be relied upon at all times to give correct results, with the least amount of labor and anxiety to the operator.

CHAPTER IX.

CARE AND USE OF INDICATOR.

Before attaching the indicator, open the cock and allow steam to blow through the pipes for the purpose of removing any scale or dirt that may remain in the pipe after fitting up; as it is of the greatest importance that all parts of the indicator be kept in good working order, where a close degree of accuracy is expected. The most important of these parts, are the cylinder and piston; to which especial attention should be directed as to their condition; because the accumulation of deposit, or dirt of any description between their surfaces of contact, (however minute) will produce irregularities, and distortions in the diagram, to such an extent, as will render it almost impossibe to secure the information a diagram is intended to convey; therefore it becomes very essential at frequent intervals to remove the piston from the cylinder; (which can be done by simply unscrewing the cap upon which the whole mechanism is attached, and lifting from the instrument) and thoroughly clean both, by the use of cotton cloth, waste, or some other suitable material.

This being accomplished, lubricate these parts with some good cylinder oil, and replace them again in the instrument.

The pivots or joints of the pencil movement, should also be kept clean, and oiled occasionally with some light machine oil

that will not gum or become sticky; a bottle of which usually accompanies the instruments of all makers.

This should be used sparingly as a very small amount suffices for the purpose, and any surplus should be cleaned off.

It is absolutely necessary that there be perfect freedom in the pencil mechanism; (and when not subject to the action of a spring) the pencil bar on being raised to its highest position, should from its own weight, fall with the utmost freedom to its lowest position, and this requirement is very essential in order to insure correct diagrams. A further test of its correctness may also be made by again raising the pencil bar to its extreme height, and covering (with the thumb or finger) the hole through which the steam is admitted against the indicator piston; and if the mechanism be not impeded in any way, except only by the air contained within the indicator cylinder, the pencil will descend slowly and uniformly until it reaches its lowest position. Should this be found otherwise than as stated, it may become necessary to disconnect the parts of the movements from the piston, and test each part separately until the cause of the trouble is located.

The Paper Drum also requires attention, it being in constant motion, and subject to considerable wear; consequently should be examined from time to time, and the bearing cleaned, and thoroughly lubricated, using the same light machine oil, as for the pencil movement.

With the ordinary paper on the drum the Siberian lead pencil about grade H. H. H. H. should be used in taking the diagrams; the pencil being sharpened to a fine round point with a knife or fine file.

A more satisfactory result of the tracings may be obtained by the use of a chemically prepared paper, (called metallic paper), upon the drum, and using a point made of common brass, or preferably silver wire suitably sharpened for the tracing point.

One sharpening of the metallic pencil will give good results on a large number of diagrams, and the general character of the work will be much more satisfactory than with the lead pencil. The paper should be placed upon the drum in such a manner that it will be perfectly smooth. This may best be done by first folding one end, and slipping it under the longer clip; then pass the paper around the drum and bring the loose end under the short clip; taking the two ends thus located, between the thumb and finger, and with the other hand by a slight pressure at the *top* of the card, slide the whole down the drum; the outer edge may then be folded back over the clip if needed.

Adjust the stop screw so that the pencil will bear lightly on the paper, otherwise the friction between the pencil and card will cause the diagram to be irregular, and will not represent the true action of the steam within the cylinder.

Paper Drum Motion. The motion of the paper drum may be derived from various different parts of the engine, but whatever point is selected, it is essential that the motion of the drum shall coincide in miniature, exactly with that of the engine piston, at every part of the stroke.

Owing to the irregular motion of the engine piston, during the stroke, caused by the varying angularity of the connecting rod, it is often uncertain, and difficult to select a point (other than the engine cross-head), that shall fulfill the exact conditions required; but if such other point should be chosen, careful attention as to its location becomes necessary, in order that the motion resulting therefrom, may in no wise vitiate the diagram.

The Cross-Head being directly connected to the piston rod, consequently moves at every part of the stroke, and under all circumstances in exact unison with the engine piston; therefore, it is the part usually selected from which to obtain the motion of the drum; as being the most direct, reliable, and

convenient for the purpose; but the amount of its movement, (whatever that may be), must be reduced to suit the range of the paper drum, or the length of the diagram to be taken; and this reduction must be in an exact proportion, throughout the stroke, to the movement of the cross-head.

Where special reducing wheels are used for this purpose, a stud is generally screwed in the cross-head, of sufficient length, such as will cause the cord from the reducing wheel, when attached, to be in a direct and parallel line with its motion.

It often happens that this plan cannot always be successfully accomplished, owing to the various different construction of engines; hence in such cases a resort to the use of small carrying pulleys will be necessary.

Carrying Pulleys. It is always desirable to avoid these pulleys wherever possible, as their use are often detrimental; in that they increase the tension on the cord; cause the cord to become dirty from the oil used in lubricating the pulley; which will in a short time render it unfit for further use; they also create a friction that becomes an additional tax upon the drum spring, and in many ways becomes a source of annoyance to the operator.

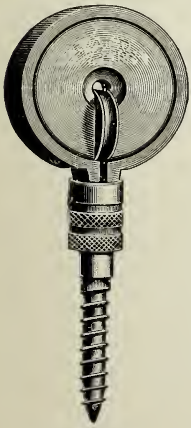


Fig. 42.

Nevertheless occasions often occur, where their use becomes indispensable, and in such cases they should be located at such points, as will suggest themselves to the ingenuity of the engineer, as best, for obtaining the desired results. The style shown in Fig. 42 is universal and meets all requirements.

Various devices, and methods for reducing the amount of movement of the engine cross-head to any desired length of diagram, will be found represented aud described in Chapters 5, 6 and 7.

The Indicator Cord. The usual manner of imparting motion from the engine cross-head to the drum, is through the medium of a hard braided linen cord; and sometimes a metallic cord (such as fine piano wire), may be used to advantage in connection with it, under circumstances where an unusual length of cord is required.

Cord of all kinds especially when new, is possessed of a certain amount of elasticity, of which it becomes necessary to remove as near as possible, before using, in order to avoid any further stretch when in use; so that coincidence of motion between the cross-head, and the reduced motion of the paper drum shall be practically uniform.

The most simple and ready method, and the plan usually adopted for removing this elasticity is by suspending the cord from one end, and attaching sufficient weights to the other; allowing it to remain so suspended from 12 to 24 hours; or until it becomes apparent that any further stretching may result in injury to the cord.

The cord that is intended especially for indicator work is usually stretched by the manufacturers until its elasticity is eliminated, and therefore will not stretch, when in use under ordinary circumstances, to any extent that would seriously interfere with the accuracy of the diagrams.

A simple, light, and exceedingly convenient cord adjuster is represented in Fig. 12, Chapter IV, for adjusting the length of the cord between the indicator and reducing gear.

The hook on the cord from the indicator, should be attached as close to the indicator as possible; (to prevent any swaying of the cord), and connect into the ring of the cord adjuster, while the small holes in the adjuster receive the cord from wherever the motion is derived.

The Indicator Spring. The denomination of the spring to be used in any particular case, depends principally upon the boiler pressure. A book usually accompanies most indicators,

which gives the necessary information, (either by a table direct, or by a computation rule), for the selection of the most suitable denomination of spring, for any given boiler pressure.

In indicators which have a range of pencil movement of from 2½ to 3 inches in height, a 40 pound spring is a good standard for pressures between 80 and 90 pounds, and a 50 pound spring for boiler pressures from 100 to 120 pounds per square inch. Each indicator spring is numbered to correspond with the pressure per square inch required to compress it an extent, sufficient to cause a vertical movement of the pencil of exactly one inch.

For example: In case a 50 spring, as shown in Fig. 13, Chapter IV, is used, a pressure of 50 pounds per square inch in the engine cylinder, will raise the pencil one inch, or a pressure of one pound will raise the pencil $\frac{1}{50}$ of an inch; the same rule applying to all other denomination of springs. After use, the spring ought never be allowed to remain in the instrument, but should at once be removed, and wiped perfectly dry; otherwise it is liable in a very short time to become corroded and pitted to such an extent, as to render it quite inaccurate; at the same time the inside of the indicator cylinder and piston, should also before laying away be made perfectly clean, and free from all moisture arising from condensed steam.

Clean by means of cotton waste, or cloth, used in connection with a wooden stick.

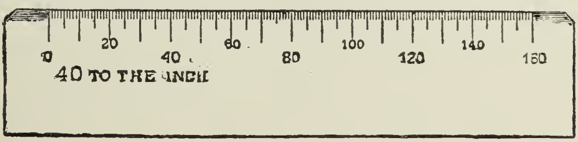


Fig. 43.

Indicator Scales. For convenience and greatly facilitating the measuring of the diagrams, special boxwood scales, as shown in Fig. 43, are provided, upon which the inches are

divided into units; each unit representing pounds pressure per square inch, corresponding to the number of the spring in use.

The edge upon which the graduations are performed, is beviled, in order that the marking may be near the paper, and consequently be more readily observed; hence their use will be found much more convenient, than the steel scales sometimes in use.

The Drum Spring. The tension on the drum spring illustrated in Fig. 44, is a matter that depends upon the good judgment of the engineer, and should be just sufficient to keep the cord taut on the backward or inner stroke of the engine. The best results are obtained where the highest tension is employed, consistent with good work. As the speed of the engine is increased, it is sometimes necessary to increase the tension on the drum spring to counteract the effect due to the inertia of the drum at the higher speed, and provisions are usually made on most indicators, and instructions given whereby the tension on the spring may be varied to suit the higher speeds wherever necessary.

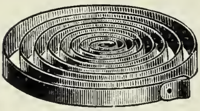


Fig. 44.

CHAPTER X.

TO TAKE DIAGRAMS.

The desired pressure spring being placed in the instrument, and the paper in place upon the drum, connect the cord from the indicator to whatever form of reducing motion there may be at hand, (thereby starting the drum in motion), then open the cock on the pipe communicating between the steam engine cylinder and the indicator piston, (which in turn starts the pencil movement) until a few revolutions of the engine have been made, or until the instrument is heated to a temperature due to the steam pressure present. During this motion, examine the pencil mechanism to see that it is moving freely, which may be ascertained by placing the finger directly against and over the top of the piston rod, and by following its motion up and down; the presence of any grit between the indicator cylinder and its piston, can readily be detected. If such should be found to be the case it will be necessary to stop the motion of the indicator, and remove the pencil mechanism with the piston and thoroughly clean and oil, before again replacing in the indicator. Again set the Indicator in motion, and after a few revolutions of the engine have been made, swing the pencil until it comes in contact with the paper on the drum and hold it there during one complete revolution of the engine. Withdraw the pencil and close the indicator cock, and immediately return the pencil to the paper in order to trace the atmospheric line. This should always be done as soon as possible after tracing the card, so that

it may be drawn under about the same conditions (as to temperature, etc.) as when the card was taken. When power is to be measured, it is a good plan to keep the pencil in contact with the paper during a number of revolutions of the engine; and in measuring the diagram, to take a line most nearly representing the average. Diagrams should always be taken from both ends of the cylinder where correct conclusions are expected.

Although a diagram from one end of the cylinder may prove satisfactory, it is not safe to infer that one from the opposite end will be equally so; but on the contrary there will often be found a great difference between diagrams taken from each end of the cylinder, owing to the varying conditions of pressure, etc., usually found in practice. Very often this difference results from improper or uneven valve setting; wherein the period of opening or closing the valve, and also the point of cut-off differ at each end in relation to the stroke of the engine; and may be partly due to rough and tortuous steam passages.

Sometimes the load may suddenly change during the interval between the taking of the first and second diagram, causing a disparity between them that might prove misleading, in that, it would give the appearance similar to that of an uneven valve adjustment, consequently after a careful consideration from a number of diagrams, it becomes a requisite of the engineer to use his best judgment in deciding the cause and applying the remedy for any irregular or unusal appearance of the diagram. This information can only be derived from a careful application of the indicator and a study of the diagrams.

Where two indicators are used and placed at each end of the cylinder, the diagrams (if desired) may readily be taken simultaneously; but in case of only one indicator at hand, it must be changed from one end of the cylinder to the other;

in order to obtain diagrams from both, a matter requiring considerable time and trouble; therefore it will be found most convenient in such case to place the indicator in connection with a three-way cock (previously described), at the middle of the cylinder, thereby admitting steam from each end, so that diagrams can be taken from either end of the cylinder, by simply turning the handle of the three-way cock to the required position. This arrangement greatly facilitates the labor, and it also enables the operator to produce diagrams from each end of the cylinder, upon the same sheet of paper, and in the shortest possible time; it being very essential that the second diagram be taken as quickly as possible after the first, in order that the conditions of speed, load, and pressure may remain more nearly alike during the time occupied for both tracings.

It is also more satisfactory to the engineer to take diagrams from both ends of the cylinder, upon the same paper as it enables him *at once* to make a discernable comparison of the pressures exerted on the opposite sides of the piston, throughout one revolution of the engine. After all required adjustments, that become necessary have been made, and a satisfactory diagram obtained, stop the motion of the drum and remove the paper therefrom. Make memorandum on the paper of as many of the facts, as may be required, such as the style of engine, where located, the diameter of cylinder, length of stroke, diameter of piston rod, the number of revolutions per minute, which end of cylinder, the scale of the spring, the vacuum in the condenser, the boiler pressure, the day and hour of taking the diagram, and any other factors that may enter into, and become necessary for an accurate consideration and solution of the diagrams.

For convenience of writing in the data, and also for filing away for future reference, the paper used upon the drum is usually in the form of printed blanks of suitable dimensions to fit the drum of the indicator, for which they are designed, and

contain the heading of the different principal factors necessary for computing from the diagram, the horse power, water consumption, etc., of the engine. For very expert tests, these

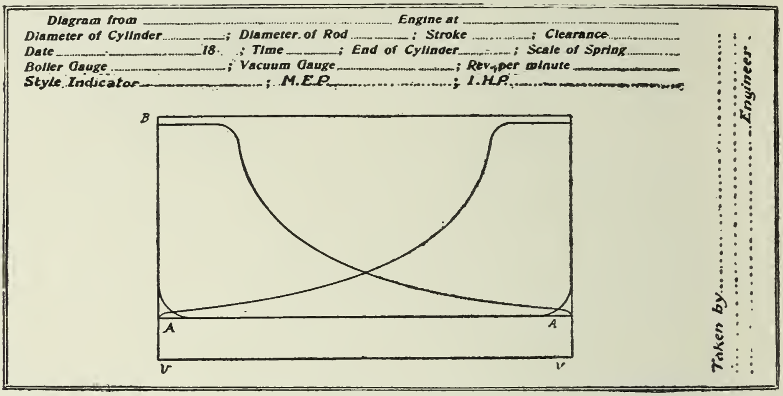


FIG. 45.

blanks may be printed in various forms, to suit any required data necessary; but the above as shown in Fig. 45 will be found sufficiently elaborate for any ordinary indicator practice.

CHAPTER XI.

INDICATOR DIAGRAMS.

The important and essential knowledge to be derived from a careful investigation and study of indicator diagrams is invaluable to the engineer, as they enable him to easily ascertain and establish various facts concerning the use of steam, that by any other method would prove complicated and unsatisfactory; of which the following may be stated.

First. It shows whether the valves of an engine are correctly and evenly timed; and also serves as a guide in all necessary adjustments of the same that may be required, in order to insure the best distribution of the steam working within the cylinder; and thereby securing the maximum economy, and efficiency of the engine.

Second. The indicator power developed in the cylinder of an engine, may be determined; also the quantity of power lost in various ways; such as leakage of valves, back pressure, too early release, and incorrect adjustment of valves.

Third. It indicates whether the steam ports, and passages are adequate in size, and a diagram taken from the steam chest, will also show whether the steam pipe and its connections are of sufficient size.

Fourth. It indicates the condition of the valves and piston in reference to leakage.

Fifth. In connection with a feed water test, (showing the actual amount of steam consumed) the economy with which the engine works may be determined.

To ascertain with accuracy, each and every item of information here mentioned, it is absolutely essential that the diagram should truly represent the motion of the piston; and also the pressure exerted on both sides of it, at every point of its stroke.

The general features of a diagram, that indicates a proper distribution of the steam in an engine cylinder, is represented in Fig. 46, the attainment of which, (as near as possible) should be the endeavor of an engineer in setting the valves of

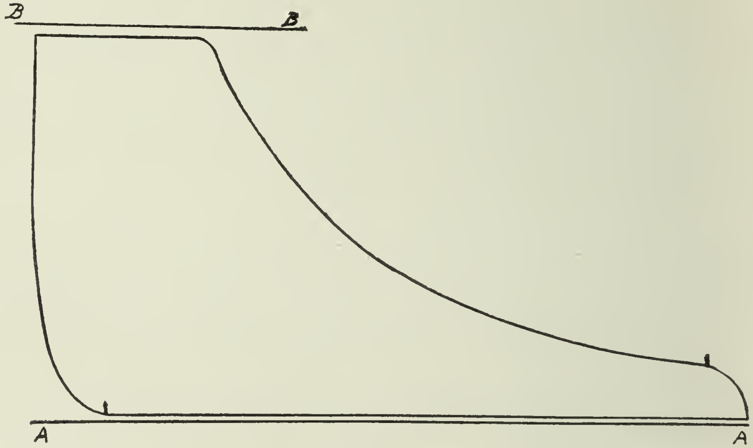


Fig. 46.

his engine. A. A. is the atmosphere line, and B. B. representing boiler pressure.

In this diagram the initial steam pressure, which is the highest pressure realized in the cylinder, is fully maintained up to the commencement of cut-off; indicating ample size of steam pipes, ports, and other passages in the engine.

The expansion curve is good, and the release of the steam is sufficiently early to secure a free exhaust, also low, and uniform back pressure.

The exhaust valve closes on the return stroke, in time to provide the necessary compression, (or cushion), and thereby

counteracting in part the effects of enertia and momentum of the piston, cross head, and other reciprocating parts, at the end of the stroke.

The admission of steam takes place promptly; and projects the admission line to initial pressure at right angles (or perpendicular) to the atmospheric line.

These qualities in a diagram being an especial requisite under any circumstances, to insure an economical working engine.

In practice however, there will be a great difference in the outline and appearance of the cards from different engines, and even from the same engine; arising from numerous circumstances and conditions connected with it.

The diagram as before stated simply shows the pressure of steam existing in the cylinder at each part of the revolution of the engine, and it is the province of the engineer to determine whether these pressures at each and every point are the correct ones; and if such is not the case to ascertain wherein the fault lies, that causes the error; then determine upon, and apply the remedy.

It must be understood, that in a great majority of cases, the shape or outline of the diagram, depends principally upon the manner in which the steam is admitted to, and released from the engine cylinder.

Therefore, by careful investigation and measurement of these outlines, and turning the varied information which they furnish to practical advantage, the real value of the indicator is readily made apparent.

As a preliminary to the study of the diagrams, suppose we knew that at a certain part of the stroke the full boiler pressure should be realized; now if this does not appear to be the case on the diagram there is evidently imperfections existing, either from an incorrect adjustment of the valves, or may be due to inadequate capacity of the steam pipes, and passages between

the boiler and engine cylinder; and almost invariably happens also with engines having insufficient, or extremely light loads.

Adversely, the diagram may show *too great* a pressure, at other certain points, when we know that there should be *less* in order that the demands for good economy and efficiency in the engine be obtained.

This latter circumstance may also proceed partly from incorrect valve adjustment; although it is principally caused by leakage through the admission valves after cut-off; in combination with the re-evaporation of steam previously condensed within the cylinder in the early part of the stroke.

Any derangement of valve mechanism of the engine, such as incorrect position of the eccentric, on the shaft or an uneven adjustment in the length of the valve rods and connections, will in consequence, be revealed in the diagram, by late admission or release, by low initial, or high back pressure, also by absence of compression; either of which in performing an equal amount of work will result in an increased consumption of steam.

Consequently where discrepancies of any kind occur, a thorough investigation, study, and reasoning of the diagram first becomes necessary, in order to intelligently locate the cause of the defect, and make changes, and corrections accordingly until the diagram shows a proper distribution of steam pressure throughout the stroke of the engine.

CHAPTER XII.

STUDY OF DIAGRAMS.

To the Steam Engine Indicator (it may be said), belongs the credit of furnishing a great part of the information that has enabled scientific engineers, (and others who have studied the subject) to apply intelligently and successfully, and has hereby contributed in a great measure to the present perfection of our modern steam engines.

The most correct and best means of obtaining a knowledge of the internal workings of the steam in a cylinder under different circumstances of loads and pressures is, by a careful study of what are designated as Indicator Diagrams.

A diagram from one end of a steam engine cylinder shows the exact pressure acting upon the engine piston, at any and every part of its movement, during the time of one complete revolution on both forward and return stroke of the engine; and to show a corresponding pressure on the other side of the piston, another diagram must be taken from the opposite end of the cylinder.

The outline of the figure traced by the pencil upon the paper of the indicator drum, depends upon the variable pressure of steam in the cylinder, acting upon the engine piston, throughout one complete revolution of the engine, in combination with the horizontal length of the figure produced by the rotative motion of the paper drum.

If the steam be admitted to the cylinder at the commencement of the stroke and continued at a uniform pressure, and exhausted at the extreme end, and from thence returning to its beginning, it will have traced a figure on the paper, bearing a close approximation to a parallelogram, or rectangle.

If the admission of steam had been cut off, after only a part of the stroke had been completed, (leaving the amount to expand to the end of the stroke) the diagram will assume a shape somewhat similar to that represented in Fig. 47.

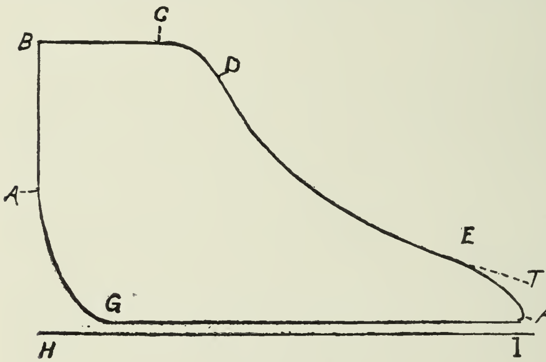


Fig. 47.

In the matter of details these two representative forms may have innumerable modifications.

We have selected Fig. 47 as exhibiting the essential features and events (during the stroke of the engine), of a well proportioned diagram, showing the action and pressures of steam that usually take place in the engine cylinder. This diagram shows that the admission of steam commences at A and continues to the point C, where cut-off commences, and is complete at D; the expansion is from D to E; the exhaust begins to open at E and closes at G; the back pressure is represented by the line from F to G, and continuing to A. The compression of the steam remaining after the exhaust closes, begins at the point G, and ends at the admission line A, B; this compression producing a continual increasing back pressure to the point A.

The point T is where the expansion line would have reached provided the exhaust had remained closed to the end of the stroke, and is designated the Terminal Pressure.

The line H, I, represents the atmospheric line; and the different events of the stroke, appearing on the diagram at various heights above the atmospheric line are measured from that line, in pounds pressure by a scale corresponding with the spring used, in producing the diagram; and consequently the location of these points in reference to each other, becomes an index to the engineer, and serves as a guidance to him, in the study of engine performance and in the steam economy of engines.

It is advisable in most cases to take the diagrams from both ends of the cylinder, on the same sheet of paper, as shown in Fig. 48, (by the use of the three-way cock, as recommended in a former chapter) in order to facilitate the matter of making a comparison between the diagrams from each end of the cylinder, more readily, and thus showing the varying pressure at different events of the stroke, in both ends of the cylinder; such as admission of steam, point of cut-off, release, or opening of exhaust, closure of exhaust, compression, etc., and thereby showing any errors or discrepancies in the adjustment of the valve, or valves, that regulates and controls the flow of steam to and from each end of the steam cylinder, and which thereby enables the engineer to arrive at correct conclusions in reference to faulty valve motions and methods of correcting them; which by any other way is a difficult task to accomplish, with *any certainty*, that the requirements of a correct valve motion are attained.

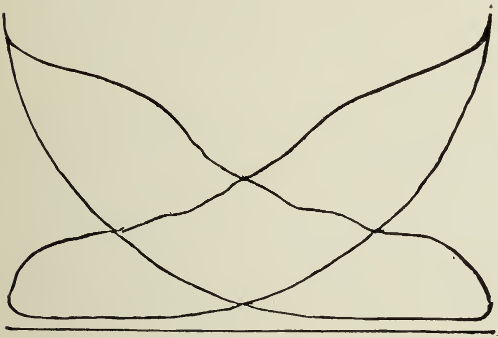


Fig. 48.

Reproduction of an actual card taken from a 9½ in. x 9 Westinghouse engine—325 revolutions per minute.

An Indicator Diagram is the result of two movements, which are at right angles to each other; one of which is the rotation of the paper drum, forward and back, around its central stud, and is produced on a reduced scale, coincident with and by the movement of the engine cross-head, and thereby tracing a horizontal line on the paper drum, at any time a contact is made between the drum and pencil.

The other is the vertical movement of the pencil, parallel to the axis of the drum and is produced by the steam pressure, acting on the piston of the indicator, and forcing it to a height proportionate to the pressure upon the piston; consequently the length of the diagram represents the stroke of the engine on a reduced scale; while the vertical height at any given point, represents the pressure upon the Indicator piston, at a corresponding point in the stroke of the engine.

The height to which the pencil will ascend, depends entirely on the pressure exerted upon the piston, and the denomination of the spring used, and is measured in pounds pressure per square inch, at any given point in the length of the diagram, by a scale, or rule, divided in a number of units per lineal inch, to correspond with the denomination of the spring.

The denomination or number of any particular spring, is one that requires the *same number* of pounds pressure per square inch on the indicator piston, to compel the pencil to move one inch in vertical height, against the resistance of said spring.

By placing a spring in the indicator and connecting the cord so as to give motion to the paper drum, (and before admitting steam to the instrument) a horizontal line may be drawn, by bringing the pencil in contact with the paper on the drum.

This line is called the atmospheric line, and from which, as a zero line, all pressures are measured in a vertical direction from the same, whether above or below the line. All

measurements above the atmospheric line represents positive pressure, while below the line, show negative pressure.

For each indicator spring, there is usually provided a special scale, or rule, for the different denominations. For example: A diagram that has been taken where a No. 40 spring was used in the indicator, (designated a 40 pound spring) should be measured by a scale that is divided in 40 parts to each lineal inch; each division in vertical height, above the atmospheric line representing one pound pressure per square inch on the indicator piston.

In the various computations and study of diagrams, for the purpose of ascertaining the Mean Effective Pressure on the Engine Piston; Horse Power; Steam Consumption, &c., and also for plotting the hyperbolic curve, it becomes necessary to establish what is called the vacuum line, or line of no pressure, which should be located parallel with the atmospheric line, and at a distance below it, equal to 14.7 pounds, by the scale corresponding with the scale of the spring with which the diagram was taken.

Another and important factor pertaining to the same study, is the *clearance line*, which is drawn perpendicular to the atmospheric line and located at the steam admission end of the diagram; and at such distance from it, as will bear the same proportion to the length of the diagram, as the volume of the clearance space bears to the piston displacement; or that volume which is equal to the area of the steam cylinder multiplied by the stroke of the piston.

The *clearance* is the amount of *waste room* between the steam valve and the engine piston when at its extreme end of the stroke, and which has to be filled with steam at initial pressure at each end of the cylinder, for every revolution of the engine.

The required amount of steam for this purpose comes *either* directly from the boiler, or, by the compression of the

steam that remains in the cylinder after an early closure of the exhaust valve, or from both combined. The amount of clearance varies in different styles of engines. In slow running engines, this variation usually amounts to from two to five per cent., whereas for high speeds it may reach from two to ten or twelve per cent., or even more.

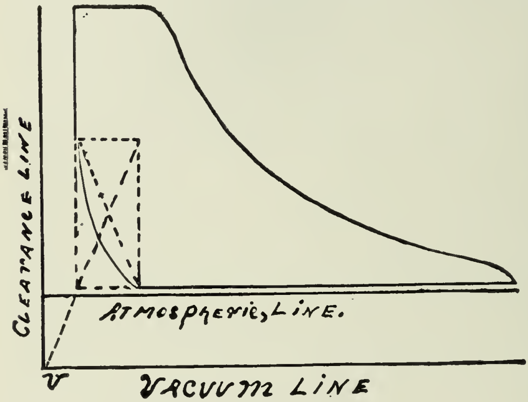


Fig. 49.

The finding of the exact amount of the clearance (in the absence of any data from the manufacturer of the engine), is often a difficult matter, but may be roughly approximated in most cases, either by computation from measurements of the space between the valve and piston, when at its end of stroke; or it may be accurately determined, (where the valves and piston are tight) by filling the space *with water* from a receptacle containing a quantity that has been previously weighed or measured, and its volume ascertained. In many cases however this is not practicable,

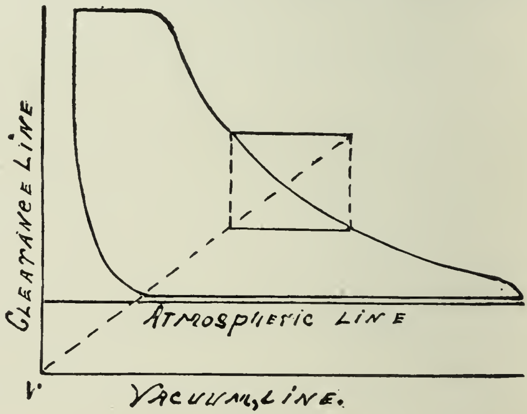


Fig. 50.

and is almost impossible, with engines that have been in use and neglected until the valves and piston become leaky, and

thereby preventing any chances of accuracy in the matter. Sometimes the only knowledge regarding the amount of clearance has to be obtained from the *diagram itself.*

If this has well defined expansion and compression curves the clearance line on the diagram may be closely determined, by a graphical method, from either curve; as shown in Figs. 49 and 50 as follows; Select two points, (preferably) on the compression curve, Fig. 49, as far apart as possible, but within the limits of the true curve, and draw a line connecting the two points, which will represent one of the diagonals of a rectangle described on the curve, of which two sides are parallel to the atmospheric line.

If now a diagonal be drawn through the opposite corners of the rectangle and extended to the vacuum line; then a perpendicular drawn from the point of intersection, will be the approximate clearance line; and the distance of this line from the end of the diagram, divided by the total length of the diagram will give the percentage of clearance in the engine.

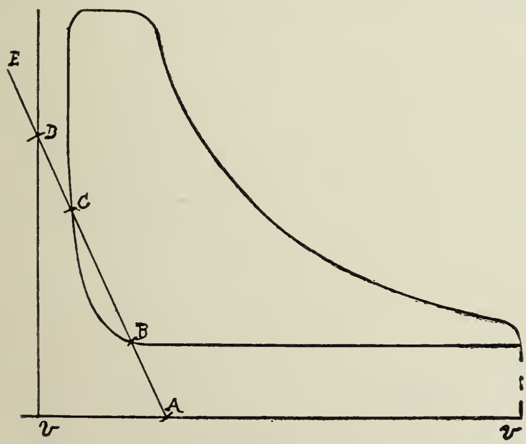


Fig. 51.

Another method of approximately ascertaining the clearance is shown in Fig. 51, the results of which coincide exactly with Fig. 49, but is given as being rather more simple in its application.

Draw the straight line A. E. from a point on the vacuum line (as at A.) in a direction as will cut the compression curve at two points B. and C. and continue it beyond the end of the diagram as at E. Now

with a pair of dividers, set one leg on the point A. and adjust the other to point B. and with this distance so taken, place one point of the dividers on point C. and describe a line intersecting the line A. E. at D.

If a perpendicular be now drawn from the vacuum line through this intersection, then its distance from the end of the diagram will represent the clearance of the engine.

The amount of clearance in an engine is an important factor, and is always considered in the study of steam economy as a source of loss; therefore in the designing of the various styles of engines, it has always been one of the principal intentions with engine designers, to construct them with a view of reducing the clearance to a minimum and thereby saving (in a measure) the amount of steam required to fill large clearance spaces.

A reduction of clearance has an effect to produce a lower terminal pressure, for any given cut-off; and also greater mean effective pressure for a given terminal; both of which are conducive to good economy.

A loss takes place from the clearance, when the steam is exhausted at a higher pressure than the back pressure, or, that pressure which exists on the return stroke and consequently a reduction of clearance, also reduces this loss of steam.

CHAPTER XIII.

LINES AND POINTS OF THE DIAGRAMS.

The diagram shown in Fig. 52, is presented in this chapter, to more fully designate by name the various lines, points, and curves, that in combination serves to make it complete, and also shows the lines dotted, that in most cases become necessary to be added by hand, in order to assist, and facilitate in all matters attending the accuracy of any calculations that may arise in reference to indicator diagrams.

The following names are generally applied to the various lines and curves of the diagram.

H. I. is the *Atmospheric Line*, and is traced by the indicator at a time when communication between the engine cylinder, and indicator piston is closed, and the atmosphere having free access to both sides of the piston of the indicator.

V. V. is the *Vacuum Line*, drawn by hand in dotted lines, and represents the line of perfect vacuum or absence of all pressure.

It is drawn $14\frac{7}{10}$ pounds, (by the scale of the spring with which the diagram is taken) below the atmospheric line; that being the mean pressure at sea level.

V. K. is the *Clearance Line*, shown drawn by hand in dotted lines, at such a distance from the end of the diagram, as will represent the total clearance or waste room between the

face of the valve, and the piston, when the engine is at either extreme end of the stroke.

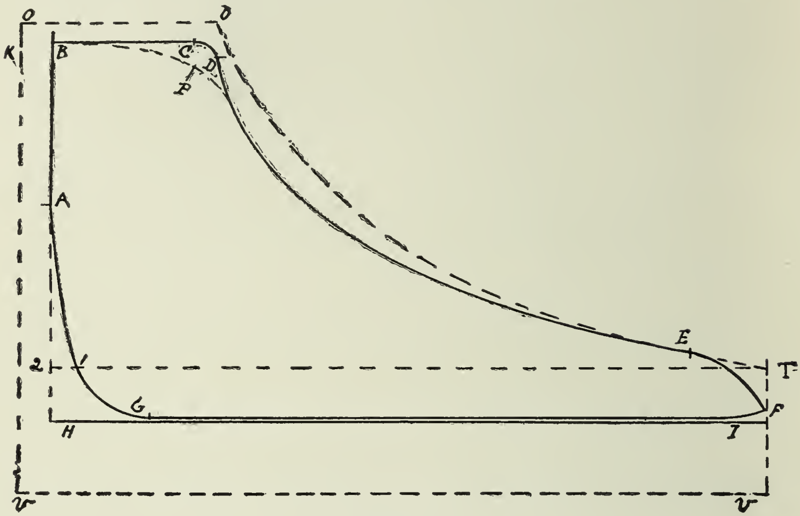


Fig. 52.

Its distance from the diagram is usually reckoned in per cent. of the piston displacement, and in Fig. 52, shows about 4 per cent. of clearance.

A. B. is the *Admission Line*, and its height above the atmospheric line represents the pressure due to the admission of steam to the engine cylinder.

It is usually very nearly perpendicular to the atmospheric line, for the reason that the admission of steam takes place very quickly, and at a time when the piston of the engine is moving very slowly, or nearly stationary.

B. is the point of *Initial Pressure* and is the first pressure realized at the beginning of the stroke of the engine.

B. C. D. is the *Steam Line*, and is traced during the time the steam is being admitted to the cylinder, or until cut-off takes place.

D. is the *Absolute Point* of cut-off and is the point where the valve closes and thereby prevents any further admission of steam to the engine cylinder.

Owing to their peculiar formations, many diagrams do not show clearly, (from observation alone) *the exact point.*

D. E. is the *Expansion Curve*, and represents the gradual fall of pressure due to the expansion of the steam remaining in the cylinder after cut-off takes place, and continuing to the end of the stroke.

E. is the *Point of Exhaust*, and is located where the exhaust valve begins to open; thereby releasing or exhausting the steam from the cylinder; and like the point of cut-off, is sometimes difficult of exact location.

E. F. is the *Exhaust Line*, which descends suddenly and is traced during the interval that occurs, between the time the exhaust valve begins to open, and the end of the stroke.

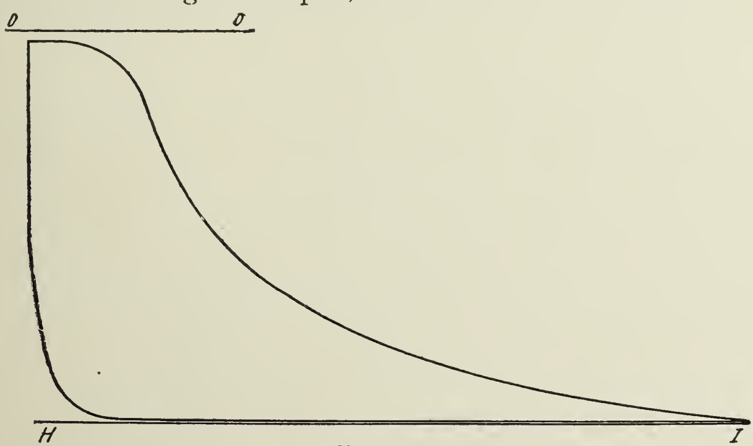


Fig. 53.

In diagrams like Fig. 53, where the pressure has gradually fallen during expansion, sufficiently low, to coincide with the return or back pressure, this line does not appear.

And in diagrams like Fig. 54, where the expansion curve falls below the back pressure, before the end of the stroke,

(causing a partial vacuum in the cylinder thereby), the exhaust line is *ascending* until it agrees with, and merges into the back pressure line.

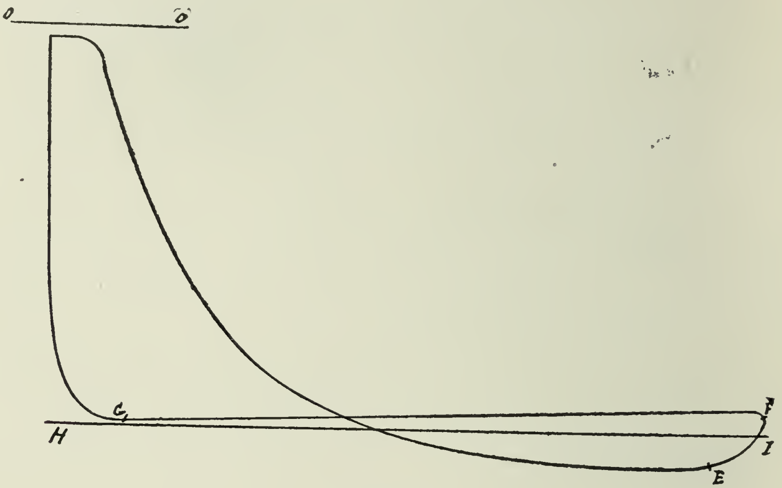


Fig. 54.

This action indicates a rapid flow of the steam from the exhaust pipe, back into the cylinder; thereby restoring the pressure lost through the expansion curve falling below the back pressure line, during the latter half of the stroke.

F. G. is the *Back Pressure Line*, and represents the pressure opposing the piston on its return movement, and for this reason is called *Back Pressure*.

In diagrams from non-condensing engines it either *coincides with*, or is *above* the atmospheric line; while in condensing engine it is *below* the atmospheric line, at such a distance as corresponds with the vacuum obtained in the engine cylinder; and in either case it acts as back pressure.

G. is the point of *Exhaust Closure*, and is where the exhaust valve closes; thereby preventing the further escape of the steam from the cylinder.

Like the point of cut-off and exhaust, it cannot in all cases be located very exactly from observation; on account of the change of pressure, due to a more or less gradual closing of the valve, which will cause its exact location to be rather undefined.

G. A. is the *Compression curve*, and is a result of the rise in pressure due to the compression (from the return motion of the piston), of the steam remaining in the cylinder, after the exhaust valve closes.

In cases where the exhaust remains open until the end of the stroke, this line does not appear on the diagram.

T. is the point of *Terminal Pressure*, and is an indispensable factor in many calculations pertaining to diagrams.

It may be located by a continuation of the expansion curve from E, to the end of the diagram at T, as shown in Fig. 52, and its height above the line of perfect vacuum, V. V. represents the absolute pressure that would exist at the end of the stroke; providing the release of the steam in the cylinder does not take place earlier.

This pressure is always measured from vacuum line V. V. hence it is the absolute *Terminal Pressure.*

Initial Expansion, is the fall in pressure that takes place through expansion during the interval between the admission of steam, and absolute point of cut-off.

It is represented in the diagram, Fig. 52, by the dotted line B. P. and is generally considered an undesirable feature, especially in automatic cut-off engines.

T. 1. and T. 2, in dotted lines, are made use of in calculations connected with the water or steam consumption of the engine, per indicated horse power, as shown by the diagram; and which will be referred to hereafter.

The Mean Effective Pressure, as before noted, is the *difference between* the average of all the varying pressures acting against the engine piston, in impelling it forward; and that of

all the average pressure which tends to retard its motion; and is another indispensable factor in the computations of the diagram; and is expressed thus: M. E. P.

The point of cut-off is an important event in the stroke of the engine, and is located at a place on the diagram, where the steam valve absolutely closes; as at D, Fig. 52, and thereby prevents the further admission of steam to the cylinder during the stroke.

With slow speed engines, and quick releasing gear, the point of cut-off will be fairly well defined; but with the higher speeds, relative to the time, or period required from the commencement, to the absolute closure of the valve, it will enable the piston to travel a certain distance during that time, (more or less according to circumstances) and which will cause the steam line to be rounded off, to meet the expansion line, and consequently the point at which the valve actually closes, can only be approximately determined by noting the point at which the curve from the steam line changes and begins to concave inward; thence continuing on, to exhaust opening, thereby forming and completing the expansion curve.

In many cases it is almost impossible to determine the point of cut-off even in this manner, owing to the various distorted formations of the lines of the diagram near this point.

CHAPTER XIV.

ISOTHERMAL CURVE.

A careful comparison and examination of the hyperbolic curve, with the expansion curve of a properly jacketed steam cylinder, with tight piston and valves, has demonstrated that the two curves conform with each other very nearly, in all respects; therefore assuming that the expansion line should be a hyperbolic curve, then the principle, upon which this curve is constructed, furnishes an easy and ready method of locating the theoretical point of cut-off, for any particular point selected on the expansion line of the actual diagram. The hyperbola, sometimes called the Isothermal curve, is constructed on the principle established by Mariotte, in reference to the compression of gases, and known as the Mariotte law; and is generally expressed as follows:

"The temperature remaining the same, the volume of a given mass of gas, is in inverse ratio to the pressure which it sustains. And this may be held to be substantially correct within a considerable range of pressure; therefore according to this law, if steam of 100 pounds absolute initial pressure, per square inch, be admitted to a steam engine cylinder (ignoring clearance in the matter), during one-half of the stroke, and admission stopped at that point; and allowing the volume at that pressure to expand during the balance of the stroke, the volume will be doubled, with a reduction of pressure, of one-half,

or 50 pounds per square inch at the end of the stroke. If, in this example, the admission of steam had been stopped at one-quarter of the length of the stroke, the volume of that amount of steam at *half-stroke* would have been doubled, but with a reduced pressure of one-half, (or 50 pounds per square inch). At three-fourth stroke, the volume would be three times, at one-third pressure (or 33-⅓ pounds), and at the end of the stroke, the volume would be increased four times, and result in a pressure of one-fourth initial (or 25 pounds per square inch), hence, the *distance* from the clearance line of a diagram, to any point on the expansion line, if multiplied by the *pressure* at such point, *the product* will be the same wherever located, and this fact furnishes a simple rule for determining any number of points through which the curve must pass, by taking the product of any point, (by such multiplication), as a constant number, and dividing it by other distances from the clearance for corresponding pressures, or by other pressures for distance from the clearance line. The properties of the hyperbola therefore enables us to locate points on the curve by an arithmetical method, described and represented in Fig. 55.

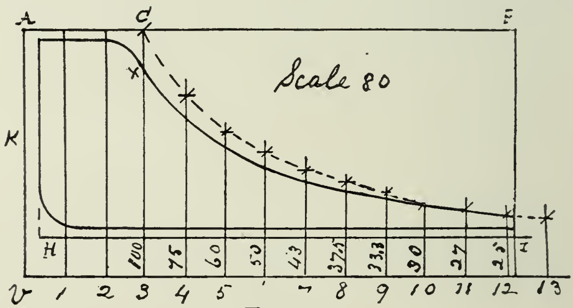


FIG. 55.

First draw the absolute vacuum, or zero line V, at a distance equal to 14.7 pounds by the scale of the spring below, and parallel, with the atmospheric line H. I. Then locate the clearance line K, in accordance with the best data at hand, in

reference to what its distance should be from the end of the diagram.

Draw A, E, to represent the boiler pressure. Select the point of cut-off on the diagram (as near as possible from observation, as at X), and draw a line through it, perpendicular to the vacuum line, and intersecting *it* at point 3, and the line A, E, at C, and this line is called the cut-off line. The point C, will then be the commencement of the hyperbolic, or theoretical curve. The vertical height of the line 3, C, (above the zero line), represents the *pressure of steam*, at the point of cut-off C; the diagram showing that pressure to be 100 pounds per square inch, (measured by the scale of the spring No. 80). The distance from the clearance line K, to cut-off line 3, C, will represent its *volume*.

Divide this volume, or distance, into any convenient number of equal parts, (it is shown divided into three parts in Fig. 55), then take the length of one such division, and (with a pair of dividers, or otherwise), commencing at the clearance line K, space on the vacuum line whatever number of these equal divisions, that may be contained in the length of the diagram, and erect perpendicular lines (a little above the actual curve), from each division. These lines are designated as ordinates, and numbered consecutively 1, 2, 3, 4, &c., beginning with the one nearest to the clearance line. It is immaterial whether the spacing comes out even with the end of the diagram; but in cases where they *do not*, it is only necessary to make an additional spacing that will extend beyond the length of the diagram, (as shown in Fig. 55), and treat it the same as the other points. Now in order to utilize the properties of the hyperbola in laying out the theoretical curve, it will be necessary to draw short lines cutting the ordinates at the proper height, measured vertically from the vacuum line; so that if the *pressure* at any ordinate, be multiplied by the *number representing the volume* of the same ordinate, the *product* will always be the same, at

whatever point selected; for example: Suppose on being measured by the scale of the diagram, (No. 80) the height of the point of cut-off C, from the vacuum line V, be found to show an absolute pressure of 100 pounds per square inch; with a *volume* equal to three of the divisions, into which the diagram has been divided. Then 100 pounds multiplied by 3, equals 300; which will be our constant number, to be divided for all other pressures or volumes. Consequently the height at which the hyperbola will cut any desired ordinate, may be found by dividing the constant 300, by the number of the ordinate; that is the height of ordinate 4, is found by dividing $300 \div 4 = 75$, and in the same manner the height of ordinate 5 is found by $300 \div 5 = 60$, or $300 \div 6 = 50$, &c., which will be the heights to be set off in divisions of the *scale* (each division representing pounds pressure), at the different ordinates. If so desired, the construction of the curve may be commenced either at the terminal pressure, or just before the point of release, and points located on the ordinates, in the opposite direction by the same method. In this case the terminal shows an absolute pressure of 25 pounds, per square inch; and having a volume of 12 divisions; therefore $25 \times 12 = 300$, which if divided by the number representing any other volume or ordinate, we have the same results for pressure as by the first method. Instead of using the height of the lines to represent pressure, they may just as well be considered to represent inches, and fractions of an inch, as follows: the vertical height of the point of cut-off C, from the zero line V, being $1\frac{1}{4}$ inches, and on ordinate 3, therefore $3 \times 1\frac{1}{4} = 3.75$ inches, which is our constant number by this method. Hence $3.75 \div 5 = .75$ of an inch, which will be the height of the point above the vacuum line, on ordinate 5, through which the curve will pass, and the heights of all other points may be found, by dividing the constant 3.75 inches, by the number representing each ordinate. The heights of all points must be measured from the vacuum line.

The location of the vacuum line may also be ascertained by dividing the [pressure of the atmosphere, 14.7 pounds, by the scale of the spring, and the quotient will be in inches; for example: $14.7 \div 80 = .183$ of an inch below the atmospheric line. This method of constructing the hyperbola, or isothermal curve, as represented in Fig. 55, is intended to show more particularly, the *principle* upon which the curve is projected, rather than to lay any claim to simplicity. There are various geometrical methods, *much more preferable*, for projecting the theoretical curve; all of which give about the same results, as the one described.

One simple and convenient plan of doing it, is represented

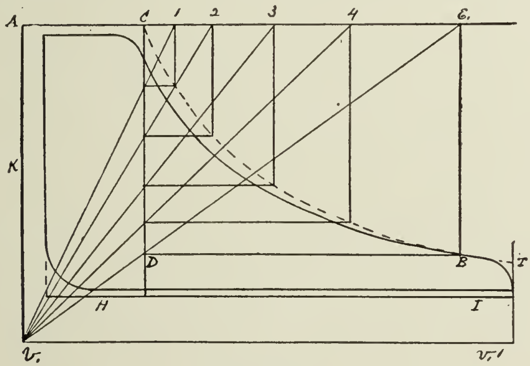


Fig. 56.

in Fig. 56, and is as follows: Draw the vacuum line V, V1, parallel with the atmospheric line H, I; at a distance below it representing 14.7 pounds, by the scale of the diagram; also draw A, E, parallel to the atmospheric line to represent the boiler pressure, erect the clearance line K, at a distance from the end of the diagram, that will represent the percentage of clearance in the engine; said line being perpendicular to and cutting the vacuum line at V. Select any point on the actual curve, before commencement of the exhaust; as at B, and from that point draw a vertical line, cutting A, E, at E. From E,

draw the diagonal E, V, and from B, draw a line parallel with the atmospheric line, intersecting the diagonal E, V, at D. From this intersection at D, erect a perpendicular cutting A, E, at the point C. Then C will be the theoretical point of cut-off, and D, C, is called the cut off line. From C, mark off any desired number of points on A, E, as 1, 2, 3, 4, &c., and draw a perpendicular from each toward the atmospheric line; also, from the same points, draw diagonals to the vacuum point V. At the intersection of the diagonals with the cut-off line D, E, draw horizontal lines to meet the perpendiculars from 1, 2, 3, 4, &c., and the intersection of these lines are the points through which the theoretical curve must pass. This method, (as well as all others), of constructing the hyperbolic curve is based on the assumption that the temperature of the steam, (or other medium) remains the same throughout its range of movement; and also that the piston and valves are absolutely tight, as well as an absence of condensation, or any other disturbing influences.

It is well known in indicator practice, that in taking diagrams from a steam engine cylinder, we are subjected (at times) to *all* of the *influences* here mentioned. The temperature of the steam changes during the stroke, and usually we find the piston and valves, more or less leaky; also initial condensation and re-evaporation takes place, (to a certain extent) all combining to cause a departure of the actual (more or less) from the true theoretical curve. Therefore one of the objects, in constructing the theoretical curve, is for the purpose of comparing and ascertaining the extent of this departure, where principally located; to *study* and *find* the *cause* of any discrepancies, so as to enable the engineer to apply the necessary means to correct, as nearly as possible, any disagreements that may appear in the actual curve.

In the construction of the theoretical curve, it has been assumed that the temperature of the steam remains the same

throughout the stroke; whereas the temperature of the steam in an engine cylinder, gradually decreases from point of cut-off to the end of expansion. Hence, (all other conditions being perfect) *temperature alone* would result in a slight disagreement, and cause the *actual curve* at its terminal (with a given cut-off) to be a little below the theoretical. The re-evaporation of the steam condensed in the earlier part of the stroke, however, will later on, tend in a measure, to increase the pressure, thereby raising the actual expansion line to more nearly conform with the theoretical curve; therefore as a general thing, a close approximation of the actual expansion curve, with the theoretical, may be taken as evidence of correct valve adjustment, and good practice. It is always advantageous to draw the true theoretical line on the diagram, in order that the actual line may be compared with it.

CHAPTER XV.

ADIABATIC CURVE AND POINT OF CUT-OFF.

The curve formed in accordance with the principles of the Mariotte law, depends for its correctness upon the condition that the temperature of the steam in the cylinder remains the same during the entire stroke; and the curve that coincides with this law of expansion, is the Hyperbolic or Isothermal, in which it is assumed that the steam within a cylinder during expansion is of exactly the same temperature throughout the length of the diagram; whereas the pressure from the point of cut-off, to the end, is continually changing, and any change in the pressure of steam, is always accompanied by a change of temperature; therefore the application of this law to a diagram from a steam cylinder, would not be absolutely correct, because for any change in volume of the steam, the corresponding change that takes place in pressure, would be *more* than if the temperature had remained constant; *or more*, than of a curve constructed in accordance with the Mariotte law. A method of improving this condition of temperature, and pressure, is by means of a steam jacket surrounding the cylinder, for transmitting heat from said jacket to the steam within the cylinder during expansion; and thereby in a measure supplying the necessary heat for re-evaporation, and also for increasing the temperature, and consequently the pressure, thereby raising the actual line of diagram, in the latter part of the

stroke and thus causing it to very nearly conform to the Isothermal or theoretical curve.

In case of exposed cylinders, or where no provision is made for transmitting heat to the steam during the stroke, a curve may be drawn approximately, representing this curve of actual conditions; wherein, all changes of volume and temperature are accompanied by a change of pressure. This curve is called the Adiabatic; as shown in Fig. 57, and may be, in fact, considered the true theoretical curve, and more nearly corresponding to the actual change of pressure that takes place, during expansion in an unjacketed steam cylinder. We are not aware of an absolutely correct method of constructing the Adiabatic curve; which term means, that when steam or other medium is under either expansion or compression, no heat enters or leaves it during that time. An approximate curve may be drawn with the aid of a table of the properties of saturated steam; although in most cases the consideration of this curve is more a problem for experts, than for the average engineer.

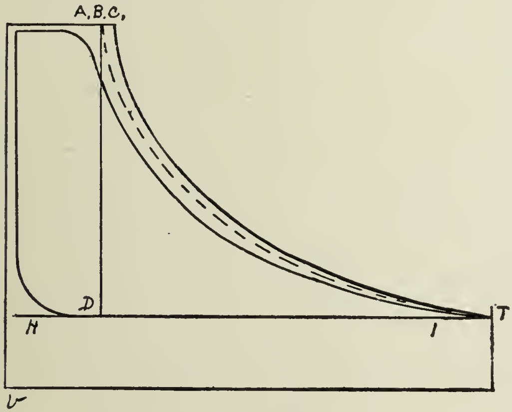


FIG. 57.

Diagram Fig. 57, is presented to show the difference between the two theoretical curves, as compared with the actual line of the diagram. The upper full line T, C, is the adiabatic, and T, B, shown in dotted lines, is the isothermal curve; while the lower full line T, A, represents the line described by the indicator. If the top, (instead of the terminal) of the curve T, C, had been made to coincide with the theoretical point of cut-off B, the adiabatic would have fallen about two

pounds per square inch *below* the theoretical curve at the terminal.

In almost all diagrams however, from engines having tight pistons and valves, with properly jacketed cylinders, and wherever a high mean effective pressure, with a low terminal is obtained, thereby securing good efficiency and economy; it is found in all such cases that the actual curve produced by the indicator, conforms very nearly with the Isothermal or theoretical curve. The fact of such agreement therefore *must* be due to a transmission of heat to the steam in the cylinder during expansion; thus increasing the temperature and thereby producing a re-evaporation of a part of the steam condensed in the earlier part of the stroke. As the isothermal curve is very easily drawn and apparently correct enough for all practical purposes, it is therefore the curve now almost universally used by all classes of engineers, for the purpose of comparing with it, the lines of the actual diagram; and where any considerable departure is found in the actual curve, all efforts are directed toward making such changes in the valve, and piston mechanism, as may be necessary to produce uniformity and a close coincidence of the actual line with the isothermal, and where they do so agree is generally considered an evidence of correct valve adjustment and efficiency of the engine. This presumption is no doubt nearly correct in most cases, with steam jacketed cylinders, and where the piston and valves are tight; but a close agreement *must not always* be taken as conclusive evidence of economical results; as we often find in practice some actual diagrams that coincide very nearly with the theoretical; but still upon investigation of the amount of steam used in the engine, they will be found to be deceptive and the opposite of economical conditions.

This deception in many cases arises from a leaky condition of the piston and valves, rather than from any lack of proper adjustment in the timing of these parts: therefore any

decided disagreement of the actual curve from isothermal, generally indicates a leakage of steam, either through the valves or piston or both; and these may enter into combination, in such a manner, as to be very misleading. The consequences of a leaky steam valve are that it will always cause the actual curve of the diagram to be higher than it should be at the terminal from a given cut-off, The actual curve in Fig. 57, represents the effect on the diagram of a leaky steam valve. In this case the Isothermal (shown in dotted lines), has been started at the end of the expansion, for the purpose of showing how much more work might have been done by the steam, from the given terminal; whereas if it had been drawn to coincide at the commencement, with the absolute point of cut-off, (that is, at the intersection of the actual curve with the line D, B,) it would have shown the expansion line of said curve considerably *higher* than the theoretical at the terminal T; this being due to the extra amount of steam entrained through the valve after cut-off and during expansion.

If, in the present instance the piston had leaked just sufficient to cause the expansion line to coincide with the theoretical; then the diagram might have been considered from *observation alone;* as representing economy and good conditions in the engine; when in fact the reverse of this is the case, owing to leaky condition of both of the parts named. Therefore in order to secure the most economical results in the steam engine, it is of the first importance, and absolutely necessary that the valves and piston, be practically tight, in order that all losses arising from this source shall be brought to a minimum. A leaky steam valve will be less noticeable at the commencement, or in the earlier part of expansion, because of the slight difference of pressure at that part of the stroke, between the steam in the cylinder and that in the steam chest, but will become more apparent on the expansion line, as these pressures become more unbalanced in the latter part of the stroke.

Also a leaky piston will be indicated by a sudden falling away of the actual curve from the theoretical at the beginning of expansion, due to the difference in pressure between the opposite sides of the piston, but as the pressures become more equalized later in the stroke, this difference will finally disappear. If the valves and piston are absolutely tight, (thereby obviating all leakage) and also, if no re-evaporation takes place, then the theoretical curve drawn strictly in accordance with the principles of the Mariotte law, *must necessarily be*, at the point of release (from a given cut-off), *higher* than the actual curve, principally on account of the increasing volume of the steam, thereby diminishing its heat, and consequently its pressure during expansion. But in indicator practice the reverse of this is usually found, and in most cases the actual curve will be above the theoretical at the point of exhaust opening. This evident rise of pressure, in the latter part of the stroke, is claimed by some engineers to be due to a re-evaporation of the steam, that has lost a portion of its heat, and therefore condensed, by contact with the colder surface of the cylinder at the commencement and earlier part of the stroke; said heat being again restored in the latter part of the stroke by transmission from the inner surface of the cylinder.

It is construed by others to be due, more to defective and leaky steam valves than to re-evaporation. It may be due either to the latter, or to a leaky steam valve, or both combined, and therefore becomes a matter for consideration and judgment in most cases on the part of the engineer.

The rise in prèssure from *re-evaporation alone*, would hardly cause the actual line to go much, if any, above the theoretical; consequently a close agreement of the actual line of the diagram to this line, is all that can be expected, or desired under the circumstances.

The location of the correct point of cut-off, on the diagram is another matter that requires considerable experience and

judgment in selecting the *best point* on the actual curve, to be used as a basis, with any of the various geometrical methods of construction, employed for locating the correct point of cut-off; all of which give about the same results.

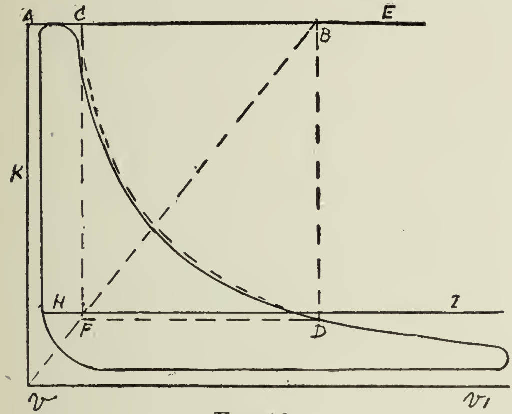


Fig. 58.

One method of construction for locating the point, is represented in Fig. 58, as follows: Draw the vacuum line V, V1, parallel and below the atmospheric H, I, at a distance equal to 14.7 pounds by the scale of the spring. Draw the line A, E, also parallel and just touching, or near the top of the diagram. The clearance being known, or determined by either of the methods previously described erect the clearance line K, accordingly; select any point on the expansion line, where it is known that the steam and exhaust valves are closed, as at D, and erect a perpendicular, intersecting the line A, E, at B; from B, draw the diagonal line to to the vacuum point V, and from D, draw a horizontal

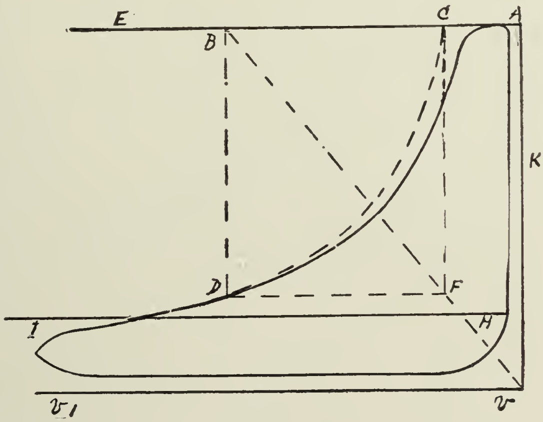


Fig. 59.

line cutting the diagonal B, V, at F. From F, erect a perpendicular intersecting the line A, E, at C. Then C, is the correct point of cut-off; or is where the further admission of steam to the cylinder must be stopped, in order that the expansion line shall pass through the selected point D. This diagram is represented as having been taken from a single cylinder condensing engine, the steam valve having no apparent leakage, and the point D, as being selected *below* the atmospheric line.

The diagram represented in Fig. 59, is taken from the same style of engine as Fig. 58 and the method of locating the point of cut-off is precisely the same. The only difference between the diagrams, being in the location of the point selected on the expansion line. Here the point D is nearer the center of the curve, and *above* the atmospheric line, and gives the correct point of cut-off at C. The construction of the theoretical curve on the diagram, in this case, indicates a considerable leak in the steam valve. It is not necessary that the line A, E, be drawn at the extreme top, as in diagrams where the valve is slow in closing, and thereby causing the curve of cut-off to be decidedly rounded, it may be better to draw the line through the absolute closure, as near as can be observed; in order that the contrast may be more plain at that point.

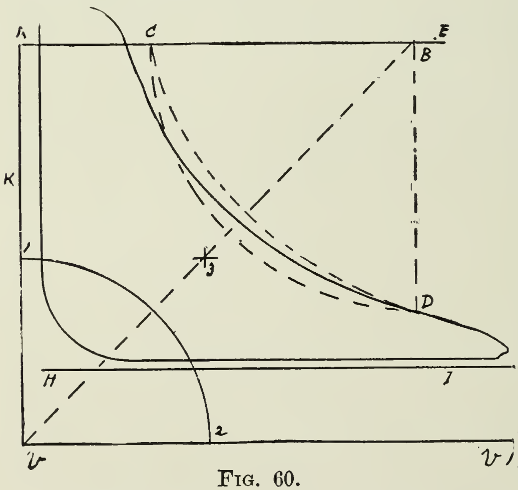


Fig. 60.

Another very simple method of locating the point of cut-off, is represented in Fig. 60, and is as follows: Draw the vacuum and clearance lines the same as in the preceding

figures; also the line A, E, near the top, or through the noted point of cut-off. From V, draw the diagonal line at an angle of 45 degrees, with the vacuum line, and intersecting the line A, E, at B, drop a line from B, cutting the expansion line at D. Place one point of a pair of dividers at B, and with a radius equal to B, D, describe the arc D, C, cutting A, E, at C, then C will be the correct cut-off point. The drawing of the diagonal line will be more quickly done by the use of a 45 degree triangle, but in the absence of one, may be done by the method as shown in Fig. 60. Place one point of the dividers at V, and with any convenient radius describe the arc 1, 2; then with the same radius, and from 1 and 2, draw short arcs, cutting each other at the point 3. From V, draw the diagonal through the intersection at 3, which will be the desired angle.

CHAPTER XVI.

THE FOOT-POUND, AND MEASUREMENT OF DIAGRAMS.

The *foot pound* is the unit of measurement in computing the power of steam engines and represents the work required to lift one pound, one foot high. The established standard of horse-power being 33,000 foot pounds or an equivalent amount of work, such as 1000 pounds lifted 33 feet; 500 pounds 66 feet; or 100 pounds lifted 330 feet in *one minute*. The horse power of a steam engine is therefore denoted by the number of pounds it is capable of raising to a given height in one minute. The usual and correct method of computing this is by multiplying the area of the piston (in square inches), by the mean effective pressure of the steam acting against the piston throughout the stroke, and also by the speed of the piston (in feet), per minute, and dividing the product of such multiplication by 33,000, the quotient will be the indicated horse-power.

For instance: Suppose we have an engine in which the piston area is 201 square inches, with a mean effective pressure of 30 pounds per square inch, and a piston speed of 450 feet per minute, then $\frac{201 \times 30 \times 450}{33{,}000} = 82.22+$ indicated horse-power. The actual efficient horse-power will be somewhat less; depending upon the amount of friction in the engine. A ready and convenient method of calculating the horse-power from a

number of cards from the same engine, is for the engineer to first compute the horse-power of his engine at one pound mean effective pressure and using the number so found as a constant or multiplier for all other mean effective pressures. For illustration: Suppose as in the preceding example, the piston area to be 201 square inches, and piston speed 450 feet per minute, then $\frac{201 \times 450}{33{,}000} = 2.74+$ which is the horse-power of the engine at one pound mean effective pressure, and which may be used as a multiplier for all other mean effective pressures in the engine; the product of the multiplication will be the total indicated horse-power. Hence the horse-power of the engine at one pound mean effective pressure as above being $2.74+$ therefore at 30 pounds it will be $30 \times 2.74 = 82.20$, or at 25 pounds mean effective pressure would be $25 \times 2.74 = 68.50$ indicated horse-power. The factors connected with the subject are therefore, as before stated. 1st. The area of the piston (in square inches), which can be obtained from a table of the diameters and areas of circles, or may be computed by multiplying the square of the diameter in inches by the decimal .7854; the product will be in square inches. 2nd. The speed of the piston (in feet), per minute and which may be found by multiplying twice the length of the stroke (in feet), by the number of revolutions of the crank, which will give the piston speed *in feet* per minute. 3rd. The force or mean effective pressure of the steam, acting upon the piston during the time. The product of all three being divided by 33,000, the standard unit of horse-power.

The indicator in most cases is used principally for determining the horse-power; but by the aid of its record made on the card, the impelling force against the piston at all periods of the stroke, is made visible, and thereby furnishes an index of the work performed, and enables the engineer to study intelligently many other important matters connected with the

problem of steam economy. In order to determine the horse-power of an engine, it first becomes necessary to ascertain from the diagram, the mean effective pressure of the steam acting upon the piston during the stroke of the engine. The finding of the mean effective pressure is rapidly and easily accomplished by the use of the Planimeter, an instrument especially adapted for the purpose; but when an instrument of this kind is not at hand, this pressure may be approximately determined by means of a number of lines drawn through the diagram as represented in Fig. 61, and is as follows: Divide the diagram into any number of equal parts, and draw lines (called ordinates), through each division and perpendicular to the atmospheric line; thus dividing the diagram into a number of small areas. The mean effective pressure may now be found by measuring the height of each line, *in pounds*, by a scale corresponding with the scale of the spring with which the diagram was taken; then by adding the pressures so found at each division and dividing their combined sum by the number of divisions into which the diagram has been divided, will give the mean effective pressure in pounds per square inch. The diagram, Fig. 61, is shown (by the ordinates in full line), to be divided into ten equal parts; consequently ten would be the divisor for the combined sum of the ordinates in this case; for example: The combined length of the ordinates measured in pounds by the scale of the

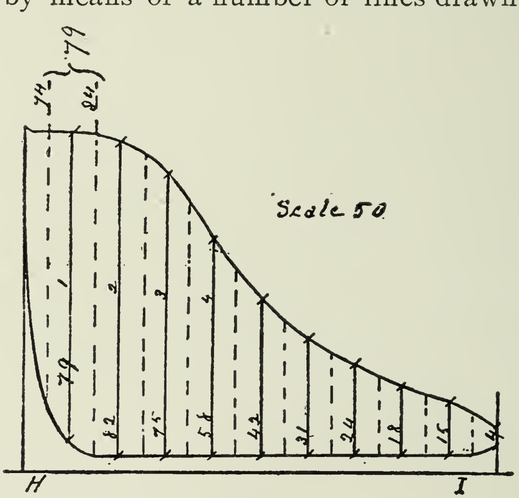


Fig. 61.

spring (50) is 428 and this divided by the number of divisions as $428 \div 10 = 42.8$ pounds mean effective pressure.

Where greater accuracy is desired, or where the outlines of the diagram are very irregular, it may be advisable to sub-divide as shown in dotted lines, making twenty divisions, and consequently dividing their combined sum by twenty. In diagrams where this irregularity exists only in a part of its length, it is sufficient (at that part alone), to sub-divide on each side of the line or lines, for which the pressure is required; and measure the pressure on each sub-division; add together and divide their sum by two (2); the quotient will be the pressure sought on the full division line as shown at the top of the diagram in Fig. 61.

In place of measuring the heights of the ordinates, (in pounds) by the scale of the spring, each may just as well be measured in inches. If the sum of their combined length be multiplied by the scale of the spring, and divided by the number of ordinates, the quotient will be the mean effective pressure in pounds per square inch, acting against the piston throughout the stroke. For example: The combined length of the ordinates measured *in inches* is 8.56, then $8.56 \times 50 \div 10 =$ 42.8 pounds mean effective pressure, the same as before. Another convenient method, which has the merit of simplicity may be staten as follows: If we draw the number of ordinates in the length of the diagram equal to the number corresponding with the denomination of the spring, then the combined length of the ordinates *in inches* will be the mean effective pressure in pounds per square inch. For example: Suppose in the diagram, Fig. 61, the spring to be a 50 and that the length of the diagram be divided into the same number (50) of equal parts, then each inch of the combined length of the ordinates would represent one pound mean effective pressure. If it had been divided into twenty-five (25) parts, then each inch would represent two (2) pounds mean effective pressure, etc., and in

any case by dividing the number representing the denomination of the spring by the number of division in the length of the diagram; the quotient will be the multiplier for the combined length of the ordinates *in inches*, and the product of such multiplication will be *in pounds* mean effective pressure. For instance: The combined length of the ten divisions in the diagram, Fig. 61, is 8.56 inches, and the scale of the spring is 50, therefore $50 \div 10 = 5$ and $8.56 \times 5 = 42.8$ pounds mean effective pressure, the result being precisely the same as by the other methods. In working out the diagrams it is advisable to make the divisions as numerous as convenient, which tends to more accurate results, and particularly so where the outline of the diagram is very irregular.

In cases where a scale of the spring is not at hand, a convenient method of finding the combined length of the ordinates is by taking a narrow strip of paper and marking on it the height of each division, commencing at No. 1. Mark its length on the paper, then place the mark made at the top of No. 1 on the bottom of No. 2 and mark the top of No. 2, and so on successively to the end of the diagram, then measuring their total combined length *in inches* which being multiplied by the number of spring, and divided by the number of divisions, will give the mean effective pressure, in pounds, per square inch, or, in place of measuring with a strip of paper, it may be as correctly done by drawing a straight line of sufficient length to contain the combined length of the ordinates, and with a pair of dividers set to the length of each different ordinate successively; commencing at No. 1, and transferring the length of each upon the line. The mean effective pressure may then be found from the total length *in inches* after all the measurements of the ordinates have been transferred to the line, by either of the two latter methods of computation.

The diagram, Fig. 62, represents the effect in a non-condensing engine of cutting off the steam at a very early part of

the stroke, and shows the expansion line crossing and running below the atmospheric line for the greater part of the stroke. This result is a general thing brought about in diagrams from an engine having insufficient load. In this case the diagram is composed of two distinct parts, and must be treated as such in computing the horse-power of the engine.

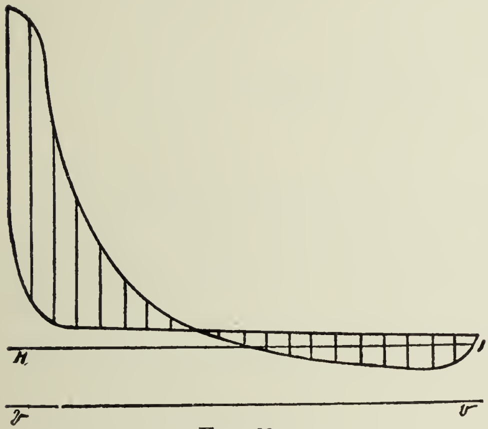


Fig. 62.

In Fig. 62, all ordinates above the atmospheric line, and to the left of where the expansion crosses the back pressure line, will represent positive pressure, and all ordinates to the right of the above, and between the back pressure and boundary line of the diagram, will be negative, and the combined length of the latter must be deducted from the former, in order to ascertain the mean effective pressure. For example: Suppose the scale of the spring to be 40, and the combined length of the ordinates of positive pressure to be 4.05 inches while the combined length of the lines of negative pressure is 1.65 inches, then the difference will be $4.05-1.65=2.4$ inches. The number of divisions of the diagram (20) being just one-half of the scale of the spring (40), it is only necessary to multiply the difference in the length of the lines by two; that is $2.4\times 2=4.8$ pounds mean effective pressure throughout the stroke. An indicator diagram like Fig. 62 always indicates a loss of efficiency and measures should at once be taken to remedy the evil.

In the matter of the computation of diagrams for mean effective pressure *alone*, it may here be well to state, that

neither the atmospheric, vacuum, or clearance lines become factors in the case; therefore we have only to deal with the lengths of ordinates *within* the actual boundary line of the diagram.

CHAPTER XVII.

EXPANSION OF STEAM.

The expansion of steam in the cylinder of an engine performing work coincides very nearly with the principle of the law pertaining to gases, and known as the **Mariotte law**; (as before noted in Chapter XIV) wherein the pressure varies inversely as the volume; the temperature remaining the same.

Upon opening communication (at any observed pressure of steam) between a boiler, and the cylinder of a steam engine, a corresponding pressure of steam will be exerted against the piston, and unless the steam be either condensed, or discharged from the cylinder, the same pressure will continue to act upon the piston; even after the valve has been closed that communicates with the boiler, and this pressure will continue so long as the volume and temperature remains unchanged.

If steam is supplied from a boiler to move a piston alternately in a cylinder, and the valve for admission of steam remains open during the full stroke of the piston, then the cylinder will be filled with steam at each stroke of the piston, of a pressure nearly equal to that of the boiler; and is consequently exhausted also at nearly the same density.

The parallelogram or rectangle shown in dotted lines b, d, v I, v, of Fig. 63 represents a *theoretical* indicator diagram from a condensing engine under such conditions; the line

v, VI being the line of absolute vacuum, and b, d, the boiler pressure.

Assume an indicator to be attached to the cylinder of such an engine, and the drum carrying the paper be given a reciprocating motion to coincide (on a reduced scale) with the motion of the engine piston: then if before admitting steam to the indicator piston, (and while the drum is in motion,) the

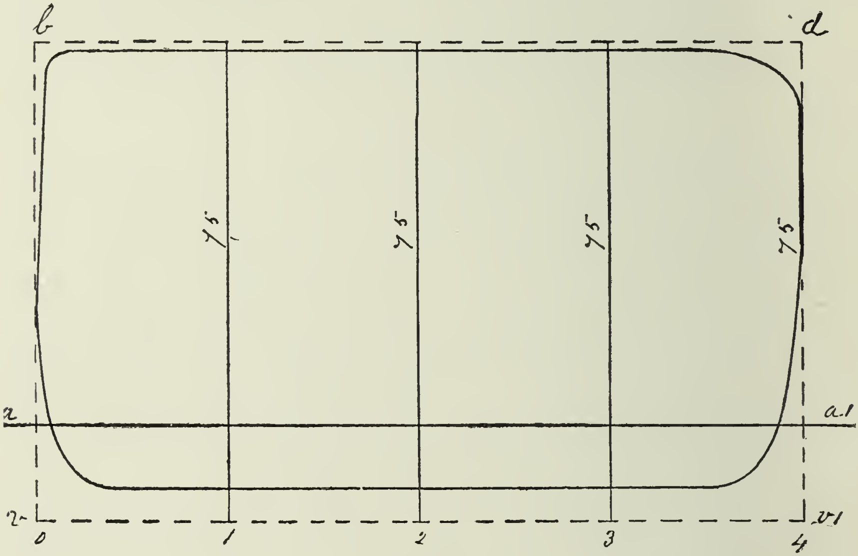


Fig. 63.

indicator pencil be brought in contact, with the drum, a horizontal line a, a1, (called the atmospheric line) will be traced upon the paper; in length, proportioned to the stroke of the engine piston; and also during this time, the pressure of the atmospheric will have free access to both sides of the indicator piston.

If communication be now suddenly opened between the steam cylinder, and indicator, at a time when the pencil is at the point a, the pencil will ascend, and trace upon the paper

the vertical line a, b, to a certain height, depending upon the pressure of the steam, and also upon the strength or scale of the indicator spring in use; and this height will represent the pressure per square inch of the steam within the engine cylinder.

Assuming now that the engine piston had just commenced to move from left to right, and the admission of steam, continues until the completion of the stroke; the pencil will have traced the line b, d, representing in this case the mean pressure of the steam throughout the stroke.

If at the point d, the valve for the admission of steam be closed, and this volume of steam be suddenly condensed by being exhausted into the condenser, (thereby creating a vacuum) the pencil will descend, and trace the line d, VI, and consequently on the return stroke follow along the line VI, V, to the commencement; thus describing a parallelogram of which the horizontal line V, VI, would represent the length of the stroke of the piston, and the vertical line v, b, would represent the total steam pressure in pounds per sqaure inch acting upon the piston; therefore the area of this parallelogram would represent pounds pressure, multiplied by the distance in feet moved through by the piston in a single stroke.

The theoretical diagram here described is one that never occurs absolutely in indicator practice; for the reason that the varying circumstances arising in the use of steam, would always preclude the possibility of obtaining such a result; therefore it is only drawn for the purpose of making a comparison of efficiency, between it, and the actual diagram, taken as near as possible under the same conditions.

In the foregoing, steam is supposed to be admitted to the cylinder, during the entire length of stroke of the piston; without any attempt to employ and utilize the benefits to be derived from the expansive properties of the steam.

The diagram in full line of the same figure, show *approximately* the outline of an actual diagram, (in practice) as the result of such an adjustment of valves, (as herein described) as would admit steam at one end, and exhaust at the other, alternately during the entire length of each stroke of the piston.

It will be observed that the actual diagram may deviate to a considerable extent, from the theoretical, in accordance with various circumstances.

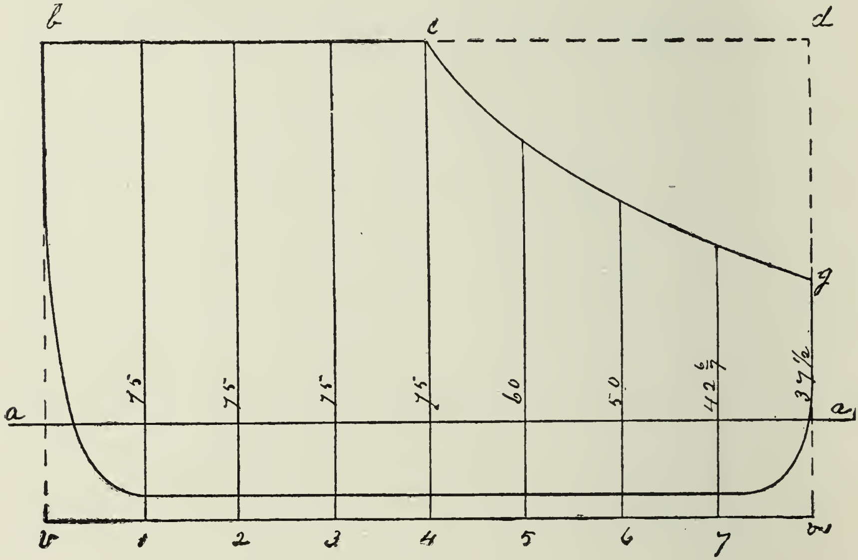


FIG. 64.

For instance, the boiler pressure may not be fully realized; also the interval of time that must elapse between the point of commencement of opening, or closing of the valves, and the absolute accomplishment of the same, will produce a wire drawing effect of the steam, and will invariably cause the corners of the diagram to be, more or less, rounded off, as shown in the diagram.

The location of the line representing the back pressure on the return stroke, will depend upon the degree of vacuum

maintained in the condenser, and this will usually be found in most diagrams, to be from three to five pounds above the line v, vi, of absolute vacuum.

The production of diagrams like the one shown in Fig. 63 are only not desirable, but the reverse of economical, and such results can only be entertained where the desire is to obtain the greatest possible power, from a given size of engine, without regard to the highest economy.

In order to save steam, or to better realize the economy, and efficiency of a given amount of steam to a greater degree, its admission to the cylinder must be stopped, or cut-off after the piston has moved *only* a *portion* of the stroke; and as the piston continues to move along the cylinder (thereby increasing the volume of the steam so confined and allowing it to act expansively), its pressure from the point of cut-off will gradually diminish to the end of the stroke, and in such proportion as corresponds to its increased volume.

Suppose we have a boiler under a steam pressure of 60 pounds per square inch, to which we add the pressure of the atmosphere (say 15 pounds), making a total of $60+15=75$ pounds per square inch *absolute pressure*.

Now if this steam be admitted to the cylinder of an engine, and the admission stopped after the piston had traveled one-half of its length of stroke, as represented by b, c, Fig. 64, it will have performed a certain amount of work, which may be represented in foot-pounds; the amount being the product of the total pressure in pounds acting upon the piston, multiplied by the distance in feet it has passed over.

If this steam, before being discharged from the cylinder is allowed to expand to double its volume, thereby forcing the piston to the end of the stroke, an additional amount of work will have been performed with this same amount of steam, and

result in effecting a decided economy in the engine; as this excess of work has been obtained through utilizing the expansion of the steam.

In this case the steam was expanded to twice its volume at the termination of the stroke of the piston, with a pressure of 37½ pounds per square inch, or just one-half what it was at half stroke.

In Fig. 64, suppose the stroke of the piston to be 4 feet and this length divided into eight equal parts, 1, 2, 3, etc., each part or volume representing six inches, or one-half foot of the stroke; then if the piston be acted upon by an absolute pressure of steam (as before stated), of 75 pounds per square inch at the beginning, and continued to the fourth division, (as at c), equal to one-half of the stroke, it will have performed an amount of work which may be represented by the mean pressure (75 pounds) multiplied by 4, $(75 \times 4) = 300$ foot pounds of work for each square inch of the area of the piston.

If the admission of steam be stopped or cut-off, after the piston has arrived at half stroke, this volume of steam in the cylinder will expand; and its pressure will gradually diminish to the end of the stroke; and the indicator pencil will trace the curve line c, g, and when the exhaust valve opens (assuming for the time a perfect vacuum in the condenser), it will descend to the point v I.

In the diagram, Fig. 64, there is only one-half as much steam admitted into the cylinder during the stroke, as in the case of diagram Fig. 63, but it will be readily observed by a comparison of the diagrams, that the area of the former is greatly in excess of half that of Fig. 63, in fact, by actual computation, (the rule for which will appear later on) its area will be found to be about .846 of that of Fig. 63, and with a mean pressure of 63.49 pounds per square inch during the entire stroke; (eight divisions) therefore the work done by the steam in the first-half of the stroke being represented by

$75 \times 4 = 300$, the amount during the whole stroke will be $63.49 \times 8 = 507.92$, hence $\frac{507.92-300}{300} = \frac{207.92}{300} = .693$, equivalent to a gain of power of about 69.3 per cent.

This has been obtained through utilizing the expansion of half the quantity of steam that would be employed during the stroke of an engine represented by the theoretical diagram Fig. 63.

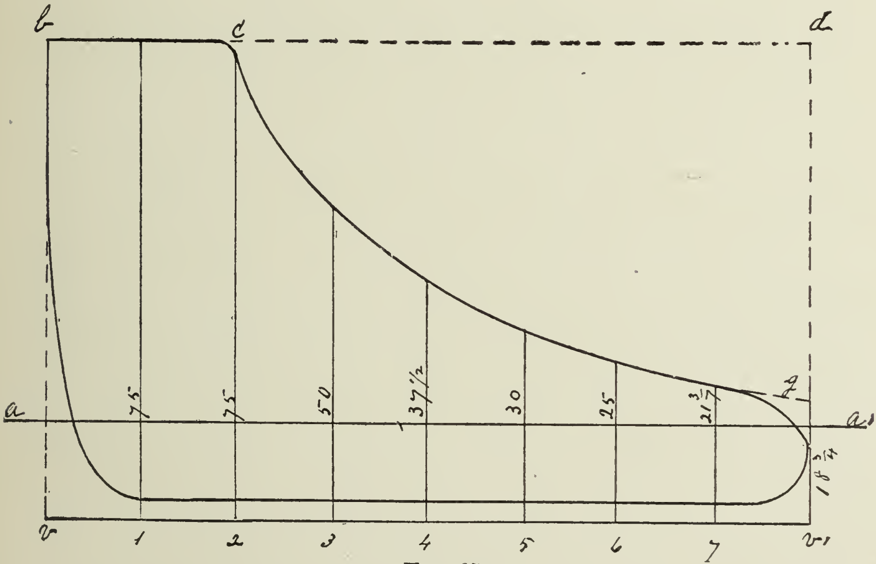


FIG. 65.

Fig. 65 is a further illustration of a diagram in which the admission of steam is cut off at one-fourth, and expanded the balance of the stroke.

In this case the amount of steam used is only one-fourth of that at full stroke, but the total area of the diagram is over .58 of the full theoretical diagram; and which is equal to a mean pressure of 44.74 pounds per square inch throughout the entire stroke, (the total initial pressure as stated being 75 pounds per square inch.)

Consequently the work done during the first quarter (one-fourth) of the stroke is represented by $75 \times 2 = 150$, and during the entire stroke by $44.74 \times 8 = 357.92$, therefore $\frac{357.92 - 150}{150}$ $\frac{207.92}{150} = 1.38$ equivalent to a gain of 138. per cent

In making calculations for pressure of steam after it has been expanded, it is the total pressure that must be considered, and which is reckoned from absolute vacuum.

Consequently the extra amount of force thus obtained, and utilized in impelling the piston the balance of the stroke, may be considered theoretically as just so much *gain*, over the single effect of the same amount of steam; as *none* of this additional pressure would have been realized upon the piston, if the stroke had terminated at the point where the steam was cut-off.

From this theoretical gain however, there are certain losses that must be deducted; such as friction of the engine during expansion; the loss of temperature caused by the gradual reduction of pressure of the expanding steam; and this loss is further increased by the abstraction of heat from the cylinder during the return stroke; thereby producing a comparatively cooling effect on the interior walls of the cylinder and also the piston.

This, as a consequence, necessitates a greater condensation of the steam, (in the earlier part of the following stroke) before the temperature of the cylinder is again restored to that of the entering or initial steam.

In practice *these losses* prevent the full theoretical economy that might be obtained; therefore in order that the maximum gain from expansion may be realized, they must be reduced to a minimum.

The usual means employed for their prevention, is by some system of cylinder covering or jacketing, also superheating; to obviate the matter of condensation; this also in

connection with the best methods of reducing the friction to a minimum.

The economy that may be derived from the expansion of steam, when used under different conditions, is an important consideration, and requires ability and good judgment (of the engineer) in arriving at the best means for realizing all expected or desired results.

The greatest gain from expansion is generally secured in Condensing Engines, but the application of a condenser however should be judiciously made, as with loads already too light, it would be of little value, and the results disappointing.

CHAPTER XVIII

HYPERBOLIC LOGARITHMS.

In the absence of an indicator, the Mean Effective Pressure of the expanding steam in a cylinder, and the power of the engine from a given pressure of steam, and point of cut-off may be approximately ascertained, and the average pressure per square inch that will be exerted against the engine piston, during the stroke can be estimated, by means of the table No. 1, of Hyperbolic Logarithms; which are calculated for expansion according to the Mariotte law.

The Hyperbolic Logarithm as found in the table, is the product of the common logarithm multiplied by 2.302585; and conversely the common logarithm is the product of the hyperbolic logarithm multiplied by 0.43429448.

The table referred to, contains the hyperbolic logarithm of numbers up to 39, which are considered sufficient for application to steam expansion.

The rule, and method of calculating the Mean Pressure by the use of the table is as follows:

Rule. To the total length of the stroke of the engine piston, (in inches) *add* the clearance in the cylinder at one end (also in inches) divide this sum by the length of the stroke at which the steam is cut-off, *added* to the *same clearance;* and the *quotient* will express the ratio *or number of expansions.*

Find in the table the logarithm of whatever *number* is nearest to that of the quotient, to which add 1.

The sum is the ratio of the gain.

Multiply the ratio thus obtained by the absolute pressure of steam *as it enters the* cylinder, and divide the product by the relative expansion; the *quotient* is the *mean pressure* required.

No.	Logarithms.	No.	Logarithms.	No.	Logarithms.	No.	Logarithms.
0.0	0.00000	4.0	1.38629	7.0	1.94591	10	2.30258
1.1	0.09530	4.1	1.41096	7.1	1.96006	11	2.39589
1.2	0.18213	4.2	1.43505	7.2	1.97406	12	2.48491
1.3	0.26234	4 3	1.45859	7.3	1.98787	13	2.56494
1.4	0.33646	4.4	1.48161	7.4	2.00149	14	2.63906
1.5	0.40505	4.5	1.50408	7.5	2.01490	15	2.70805
1.6	0.46998	4.6	1.52603	7.6	2.02816	16	2.77259
1.7	0.53063	4.7	1.54753	7.7	2.04115	17	2.83321
1.8	0.58776	4 8	1.56859	7.8	2.05415	18	2.89037
1.9	0.64181	4.9	1.58922	7.9	2.06690	19	2.94444
2.0	0.69315	5 0	1.60944	8.0	2.07944	20	2.99573
2.1	0.74190	5.1	1 62922	8.1	2.09190	21	3.04452
2.2	0.78843	5.2	1 64865	8.2	2.10418	22	3.09104
2.3	0.83287	5.3	1.66770	8.3	2.11632	23	3.13549
2.4	0.87544	5.4	1.68633	8.4	2.12830	24	3.17805
2.5	0.91629	5.5	1.70475	8.5	2.14007	25	3.21888
2.6	0.95548	5.6	1.72276	8.6	2.15082	26	3.25810
2.7	0.99323	5.7	1.74046	8.7	2.16338	27	3.29584
2.8	1.02962	5.8	1.75785	8.8	2.17482	28	3.33220
2.9	1.06473	5.9	1.77495	8.9	2.18615	29	3.36730
3.0	1.09861	6.0	1.79175	9.0	2.19722	30	3.40120
3.1	1.13140	6.1	1.80827	9.1	2.20837	31	3.43399
3.2	1.16314	6.2	1.82545	9.2	2.21932	32	3.46574
3.3	1.19594	6.3	1.84055	9.3	2.23014	33	3.49651
3.4	1.22373	6.4	1.85629	9.4	2.24085	34	3.52636
3.5	1.25276	6.5	1.87180	9.5	2.25129	35	3.55535
3.6	1 28090	6.6	1.88658	9.6	2.26191	36	3.58352
3.7	1.30834	6.7	1.90218	9.7	2.27228	37	3.61092
3.8	1.33046	6.8	1.91689	9.8	2.28255	38	3.63759
3.9	1.36099	6.9	1.93149	9.9	2.29171	39	3.66356

TABLE No. 1.

For example. Suppose steam of 100 pounds *absolute pressure* per square inch, be admitted to the cylinder of an engine, at the beginning of the stroke (ignoring clearance for the

present), and admission stopped after one-fifth (1-5) of the stroke had been completed, and the steam allowed to gradually expand to the end of the stroke.

Then in accordance with the principle of the law, in reference to the expansion of gases, the volume of steam in this case, on being continually increased, will consequently suffer a corresponding reduction in pressure.

At 2-5 of the stroke the volume will be double, and the pressure reduced to ½ of the initial, or to 50 pounds per square inch; at 3-5 to ⅓ or 33⅓ pounds; at 4-5 to ¼ or 25 pounds, and at 5-5 or the whole stroke the volume is increased 5 times, with a reduction of pressure of 1-5 initial or to 20 pounds pressure per square inch at the termination of the stroke.

Now what we desire to ascertain in such a case, is the average pressure during the entire stroke or, what pressure acting uniformally throughout the stroke will perform an equivalent amount of work.

This may be calculated very readily by the use of the Table No. 1, in connection with the Rule. Suppose in the foregoing example, the stroke of the piston to be 60 inches, and the admission of steam be stopped after the piston had advanced 12 inches and expansion continuing to the end of the stroke; then by the Rule $\frac{60}{12}=5$ the ratio of expansion.

This ratio of expansion (5) will be found in the table No. 1, under the head of *numbers*, and directly opposite (to the right) will be found its hyperbolic logarithm 1.609, to which add 1. the sum of which $1.+1.609=2.609$ the ratio of gain.

Multiply 2.609 by the initial pressure of steam entering the cylinder, and divide the product by (5) the ratio of expansion: hence $\frac{1.609\times 100}{5}=52.18$ which represents in pounds per square inch, the average or Mean Pressure that would be exerted uniformally against the piston during the entire stroke of the engine. If the stroke of the engine piston in the above

example had been 48 inches, the steam cut-off, after the piston had advanced 12 inches, and the absolute initial pressure of the entering steam be 80 pounds per square inch, then $\frac{48}{12}=4$ equal the ratio of expansion; the logarithm of which is 1.386 and represents the ratio of the gain.

Hence $\frac{1.+1.386\times 80}{4}=\frac{2.386\times 80}{4}=47.72$ which would express the Mean Pressure in pounds per square inch impelling the engine piston during the entire stroke.

In computing the above examples for the Mean Pressure, the effect of the clearance in the cylinder, has been purposely neglected, in order that the calculation might be presented in a more simple manner.

In the case of the latter example, suppose the percentage of clearance to have been such as to add two (2) inches in length to each end of the cylinder; then $48+2=50$ inches, the length of stroke with the clearance at one end added; and by adding the same clearance to the distance of cut-off, $12+2=14$ inches, therefore $\frac{50}{14}=3.57$ the ratio of expansion in this case.

From the table No. 1, we find the nearest number to this ratio of expansion is 3.55 the logarithm of which is 1.267 to which add 1. then $1.+1.267=2.267$ and $\frac{2.267\times 80}{3.57}=50.80$ which represents the Mean Pressure per square inch, when computed with the clearance included, instead of 47.72 as before; an increase of 3.08 pounds per square inch.

As all calculations of this kind are generally made for approximate results or comparisons *only*, it is in most cases unnecessary to take the clearance into consideration, unless it should be unusually large, or unless the cut-off should take place very early in the stroke; either of which, or both combined would in a measure cause a variation in the final results.

This indifference, relative to the clearance in the calculations proceeds from the fact that the full boiler pressure is never fully realized in the cylinder; and also on account of a falling of pressure that often takes place in the cylinder before cut-off; and it may be assumed that what will be gained by clearance, is about offset by the *failure of the steam* to fulfill the conditions required.

Portion of stroke at which steam is cut off.	Grade or ratio of expansion	Hyperbolic logarithm	Mean pressure of steam during the whole stroke	Percentage of gain in fuel or power
l	g	x	p	%
$\frac{1}{10}$, or 0.1	10.0	2.302	3.302	230.0
$\frac{1}{8}$, or 0.125	8.0	2.079	3.079	208.0
$\frac{1}{6}$, or 0.166	6.0	1.791	2.791	179.0
$\frac{2}{10}$, or 0.2	5.0	1.609	2.609	161.0
$\frac{1}{4}$, or 0.25	4.0	1.386	2.386	139.0
$\frac{3}{10}$, or 0.3	3.33	1.203	2.203	120.0
$\frac{1}{3}$, or 0.333	3.0	1.099	2.099	110.0
$\frac{3}{8}$, or 0.375	2.66	0.978	1.978	97.8
$\frac{4}{10}$, or 0.4	2.5	0.916	1.916	91.6
$\frac{1}{2}$, or 0.5	2.0	0.693	1.693	69.3
$\frac{6}{10}$, or 0.6	1.666	0.507	1.507	50.7
$\frac{5}{8}$, or 0.625	1.6	0.47	1.47	47.0
$\frac{2}{3}$, or 0.666	1.5	0.405	1.405	40.5
$\frac{7}{10}$, or 0.7	1.42	0.351	1.351	35.1
$\frac{3}{4}$, or 0.75	1.33	0.285	1.285	22.3
$\frac{8}{10}$, or 0.8	1.25	0.223	1.223	20.5
$\frac{7}{8}$, or 0.875	1.143	0.131	1.131	13.1
$\frac{9}{10}$, or 0.9	1.11	0.104	1.104	10.4

TABLE NO. 2.

It must be understood that the Mean Pressure as estimated in the preceeding examples, is the *absolute pressure*, measured from the line of perfect vacuum.

In non-condensing engines under good conditions, the average back pressure will be from one to two pounds above

the atmosphere, or about 16 pounds absolute; and which must be deducted from the results obtained. The remainder will be the average, or Mean Effective Pressure.

In condensing engines the back pressure will average from about 4½ to 5 pounds irrespective of the atmosphere, being a loss through imperfect vacuum; and which must also be deducted in this case, in order to obtain the Mean Effective Pressure. In either case the exact amount of back pressure to be deducted will vary; and such variation will depend mostly upon circumstances, and local conditions.

The theoretical economy of using steam expansively is given in Table No 2, which contains the hyperbolic logarithm for numbers running from 10, the grade, or ratio of expansion, representing 0.1, or 1-10 cut-off, to 1.11, representing 0.9, or 9-10 cut-off, and which may be considered of sufficient range for application to the expansion of steam in engines for all practical purposes.

The first column L represents the portion of stroke at which steam is cut-off, the second G the grade, or ratio of expansion, the third X the hyperbolic logarithm of the number or grade of expansion; the fourth, the mean pressure of steam during the whole stroke, and the fifth column the percentage of gain in power.

The per cent of gain by expansion is obtained by multiplying the *logarithm* of the number of expansions by 100.

In the Table no deductions are made for a reduction of the temperature of the steam during expansion, nor for any loss through back pressure.

In expansion, the same relative advantages occur, as given in the table whatever may be the initial pressure of the steam.

The results, in reference to the percentage of gain as shown by the table, is as before stated, theoretical, as from the

resistance to expansion of the back pressure in a cylinder, and from the loss of temperature of the steam by cooling, and also from the friction of the steam passages, these results in practice are very materially reduced.

The pressure of the atmosphere is always included in calculating the expansion; therefore must be deducted from the results in all non-condensing engines.

CONSTANTS FOR FINDING THE AVERAGE PRESSURE IN THE CYLINDER WITH ANY PRESSURE OF STEAM.

Percentage of the stroke at which steam is cut off.	Constant.	Percentage of the stroke at which steam is cut off.	Constant.	Percentage of the stroke at which steam is cut off.	Constant.	Percentage of the stroke at which steam is cut off.	Constant.
1%	·0560	21%	·5377	41%	·7758	61%	·9114
2	·0982	22	·5529	42	·7841	62	·9162
3	·1321	23	·5679	43	·7920	63	·9200
4	·1688	24	·5823	44	·8010	64	·9264
5	·1998	25	·5967	45	·8088	65	·9298
6	·2288	26	·6102	46	·8164	66	·9340
7	·2563	27	·6237	47	·8235	67	·9385
8	·2821	28	·6365	48	·8318	68	·9427
9	·3067	29	·6484	49	·8396	69	·9461
10	·3302	30	·6612	50	·8466	70	·9496
11	·3527	31	·6726	51	·8536	71	·9531
12	·3743	32	·6842	52	·8592	72	·9588
13	·3952	33	·6958	53	·8660	73	·9595
14	·4152	34	·7066	54	·8722	74	·9620
15	·4345	35	·7172	55	·8779	75	·9638
16	·4532	36	·7276	56	·8846	80	·9784
17	·4712	37	·7378	57	·8904	85	·9878
18	·4885	38	·7477	58	·8962	90	·9945
19	·5055	39	·7567	59	·9002	95	·9960
20	·5219	40	·7665	60	·9062	100	1·0000

TABLE No. 3.

In condensing engines a deduction must be made for imperfect vacuum; usually amounting to, from 2½ to 3 pounds per square inch.

The Table No. 3 contains constants for finding the average pressure in the cylinder, for any percentage of the stroke (from

1 to 100) at which the steam is cut-off, and on account of its simplicity, will be found in many cases more convenient in finding the average pressure on the piston throughout the stroke (from a given initial), than by the ordinary method of using hyperbolic logarithm.

The rule by which to use the constants is as follows: Multiply the constant opposite the known per cent of cut-off, by the total pressure of the steam entering the cylinder; the product will be the total average pressure on the piston.

AVERAGE PRESSURE OF STEAM IN THE CYLINDER WITH VARIOUS INITIAL PRESSURES AND DIFFERENT RATES OF EXPANSION.

Initial pressure above atmosphere in lbs. per sq. inch.	Percentage of the stroke at which steam is cut off.									
	10	15	20	25	30	35	40	45	50	60
	Average pressure in lbs. per square inch during the whole stroke.									
40	18·1	23·8	28·6	32·8	36·6	39·3	42·1	44·4	46·5	49·9
45	19·8	26·2	31·0	35·8	39·2	43·0	46·0	48·6	50·7	54·4
50	21·5	28·3	33·8	38·7	43·0	46·5	49·9	52·6	55·0	59·0
55	23·2	30·5	36·4	41·7	46·2	50·2	53·5	56·7	59·1	63·3
60	24·7	32·6	39·0	44·8	49·5	53·7	57·4	60·8	63·4	68·0
65	26·4	34·8	41·7	47·7	52·9	57·3	61·1	64·8	67·5	72·5
70	28·0	37·0	44·1	50·6	56·1	60·9	65·0	68·8	71·8	77·0
75	29·7	39·2	46·9	53·7	59·4	64·3	68·8	72·9	76·0	81·5
80	31·3	41·4	49·4	56·6	62·8	68·0	72·7	76·9	80·2	86·0
85	33·0	43·5	52·0	59·6	66·0	71·5	76·5	80·9	84·4	90·6
90	34·7	45·7	54·7	62·7	69·5	75·0	80·3	84·9	88·7	95·3
95	36·3	47·9	57·5	65·9	73·0	78·6	84·1	89·0	93·0	100·0
100	38·0	50·0	59·9	68·7	76·0	82·2	88·0	93·0	97·3	104·5
110	41·3	54·5	65·0	74·6	82·6	89·3	95·7	101·0	105·6	113·0
120	44·7	58·8	70·1	80·5	89·3	96·5	103·3	109·1	114·0	122·1
130	48·0	63·1	75·3	86·5	96·0	103·8	111·0	117·2	122·5	131·3
140	51·2	67·5	80·7	92·5	102·6	111·0	118·6	125·3	131·0	140·4
150	54·6	71·9	85·9	98·5	109·3	118·0	126·3	133·4	139·5	149·5
160	57·9	76·2	91·1	104·5	116·0	125·2	134·0	141·5	148·0	158·5
170	61·2	80·6	96·3	110·5	122·6	132·4	141·8	149·7	156·5	167·5
180	64·5	85·0	101·5	116·5	129·3	139·7	149·5	157·9	165·0	176·6
190	67·9	89·3	106·7	122·5	136·0	146·5	157·3	166·0	173·5	185·7
200	71·1	93·7	112·0	128·5	142·6	154·0	165·0	174·0	182·0	194·8

TABLE No. 4.

From this total, subtract the average back pressure, (which will be about *five pounds* in condensing engines; and in

non-condensing engines it will be from one to two above the atmosphere, or about sixteen pounds total), the remainder will in either case be the average, or mean effective pressure.

Table No. 4 gives the average pressure of steam in the cylinder of an engine for the various initial pressures, above the atmosphere in pounds per square inch, and at different rates of expansion.

In the above table, no allowance is made for back pressure and compression, therefore their effect must be subtracted from the above average pressures in order to ascertain the *mean effective pressure* on the piston.

In non-condensing engines, working under favorable conditions, the average back pressure will be from 1. to 2. pounds above the atmosphere or ordinarily a total of about 16 pounds per square inch; this amount varies according to location, or elevation above the sea level.

In condensing engines the back pressure will average about 5 pounds, irrespective of atmospheric pressure.

CHAPTER XIX.

THEORY OF ACTION OF STEAM EXPANSION IN CYLINDERS.

There are three conditions of the steam cylinder, within which the action of the steam is differently influenced, as follows:

1st. The outer surface may be bare or unprotected by any covering whatever and wholly exposed to the surrounding medium.

2d. It may be covered by some non-conducting material, such as felt or asbestos, and this in turn protected by a covering of wood or iron on the outside.

3d. It may be so constructed than an annular chamber may be formed on the outside, to be filled with steam from the steam chest, steam pipe, or from any convenient place where the steam is, of at least, the same temperature as the entering steam driving the piston.

In some cases the cylinder heads are also cast with a chamber for containing steam; this is called steam jacketing, and the jacket itself is also covered with a non-conducting material.

Therefore an explanation of the generally accepted theory of cylinder condensation may be of assistance to many, and lead to a better understanding of the action of steam within the cylinder of an engine while in operation, and also explain the cause of such condensation; which invariably occurs, more or less, in all steam engine cylinders.

The action of steam in an unjacketed or exposed non-condensing engine cylinder is about as follows: As the entering steam at the commencement of the stroke, is of a much higher temperature than the metal parts, with which it comes in contact; (that is, the interior surface of the cylinder, piston, and cylinder head;) consequently a portion of this steam is condensed in heating up these parts, to the temperature of the entering steam.

As the piston moves forward uncovering fresh surfaces of the cylinder, the condensation continues until after the admission valve closes, or until cut-off takes place.

This condensation is deposited in form of moisture upon the interior walls of the cylinder, also the piston and cylinder head; but does not become apparent on the steam line of the diagram, because the place of that condensed, is supplied from the steam chest, during the admission of steam.

After cut-off the steam then commences to expand, and both pressure and temperature begin to diminish in a correspondingly degree; and unless the steam is cut-off very early in the stroke but little further condensation takes place; (although fresh surfaces of the cylinder that are cooler than the steam, continued to be uncovered as the piston advances) for the reason that as soon as the temperature of the steam commences to fall, through expansion, the head, piston, and walls of the cylinder, (already heated to the temperature of the initial steam,) begins to impart a portion of their heat to the expanding steam, and thus prevent further condensation (to any great extent) taking place. As represented at A in diagram Fig. 66. As the piston continues to advance, the expansion is carried still further and in consequence, the temperature, as well as pressure, is correspondingly lowered.

During this time the higher temperature existing in the cylinder walls, piston and head, has been gradually imparted to the steam condensed in the early part of the stroke, causing

a re evaporation of this moisture and thereby raising the terminal pressure in the cylinder. Also shown in Diagram 66 at B.

When the piston arrives at or near the end of the stroke, the exhaust valve opens, and both pressure and temperature of the steam immediately falls to an extent corresponding to the pressure and temperature of steam at atmospheric or back pressure.

As the metal has still a higher temperature than the exhaust, any remaining water is therefore re-evaporated during

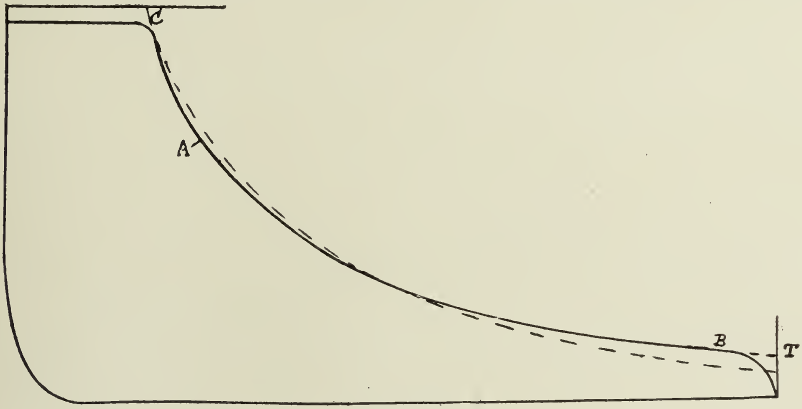


Fig. 66.

the return stroke, by absorbing heat from the cylinder walls, piston, and head, thereby still further reducing their temperature; and this extraction of heat has to be restored to these parts again at the expense of the entering steam for the next forward stroke of the engine.

In a *very early* cut-off in unjacketed cylinders the steam suffers further condensation for some distance after the admission valve closes, owing to the cooler portions of the cylinder surface which are being exposed (by the advance of the piston,) having to be heated by the *comparatively small volume of steam* confined within the cylinder after cut-off, and the consequence

is that the pressure falls for some distance in a much greater ratio, than that due to expansion alone. As the piston advances however, and the expansion continues, the pressure falls, and consequently the temperature of the steam becomes less than that of the interior surface of the cylinder, and other parts; thereby causing a *re-evaporation* of the moisture which has been deposited upon their surface during the earlier part of the stroke. The volume of steam present, being thus in-

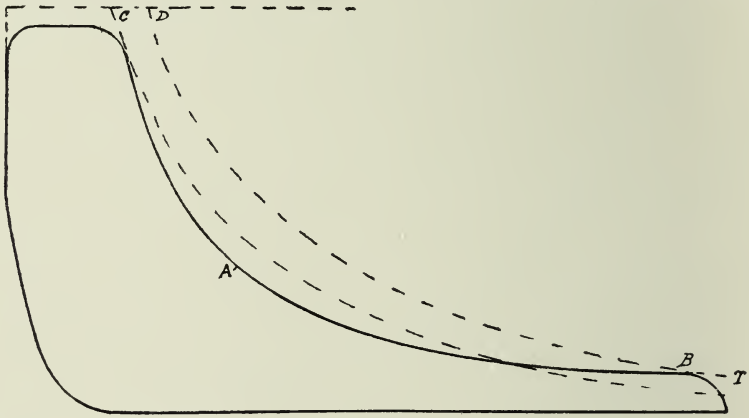


FIG 67.

creased by this re-evaporation, the pressure also becomes higher, resulting in a rise of the expansion line during the latter part of the stroke.

Therefore the effect of this action on the expansion curve of an actual diagram, with an early cut-off, is to cause it at first to *fall considerably below*, (just after cut-off) and subsequently to *rise above* the true theoretical curve toward the end of the stroke.

The expansion line of Fig. 67, represents the partial increased effects of cylinder condensation, that is due to an early cut-off, showing the falling below at a point A, and rise above at B, of the actual from the theoretical curve drawn in dotted line from the point of cut-off C.

The theoretical curve D, B, is drawn in dotted line from the point B, at which the exhaust valve opens, and represents the additional work that might be done by steam of a terminal pressure T, provided condensation were prevented.

This deviation from the true curve is greatly increased by water held in suspension or entrained in the steam.

It is therefore important that the initial steam be practically free from moisture.

The mutation of heat back and forth, (which occurs at every stroke) between the steam, and the interior surface of the cylinder as well as the cylinder head and piston, take place very quickly, and effects the metal of these parts to a slight depth only; as there is not sufficient time for it to penetrate very deeply, especially in high speed engines.

Steam Jacketed Cylinders. The action which takes place in a *steam jacketed cylinder* of a condensing engine is some what different; as the following description will indicate:

In such cases the jacket is arranged to be supplied constantly with *direct steam* (either through the steam chest, or steam pipe) of the same temperature and pressure as the initial steam entering the cylinder; consequently the alternate heating, and cooling of the metal that occurs in an unjacketed cylinder, will in a great measure in this case be prevented; hence, comparatively no initial condensation takes place, and the steam will enter the cylinder without apparent loss.

The piston itself being partially under the same conditions as before, will tend continuously to condense a very small portion of the steam; but this condensation, (in the act of forming) will at once be re-evaporated; therefore no actual condensation takes place during expansion, as the quantity of heat that disappears in doing work is steadily supplied by the cylinder walls.

The walls in turn absorb heat from the steam in the jacket, thereby condensing a portion of the steam, but the jacket being constantly supplied with direct steam, maintains the cylinder at nearly a uniform temperature.

When the exhaust valve opens, and communication is made with the condenser, (there being no water or moisture to re-evaporate,) a further expansion of the steam occurs; thereby lowering both pressure and temperature.

This exhaust steam being comparatively dry, receives and parts with heat slowly, and therefore does not absorb as much heat from the cylinder walls when expanding into the condenser, as the wet steam from an unjacketed non-condensing cylinder; as in the former case.

Although the steam jacket supplies the heat necessary to prevent condensation, and also to heat up the cylinder, from a temperature corresponding to the exhaust, to that of the initial or entering steam; yet this quantity of heat is much less than that which is extracted by wet steam from the walls of cylinders that are unjacketed, consequently the actual gain effected by the use of a steam jacket on a cylinder, is, the difference of saving, between the prevention of condensation in the cylinder during the first part of the stroke, and the loss that occurs in heating the exhaust steam during the return stroke; and this gain may in many cases be very slight, as the saving depends principally upon the fact that steam absorbs heat much slower than water.

In preventing this condensation in the cylinder, the heat abstracted from the steam jacket, transfers all liquefaction or condensation of the steam to the jacket; and on this account some engineers at the present time, questions its utility in the matter of economy, (also considering first-cost) and claim that the condensation, and waste of steam in the jacket, is more than that lost or wasted in unjacketed cylinders; the excess

being due to increased condensing, and radiating surface of the jacket, above that of the steam cylinder.

The diagram represented in Fig. 68 is from a steam jacketed cylinder and it will be seen that the actual curve agrees very closely to the Isothermal.

In reference to the efficiency of the jacket however, all results depend in a great measure upon its proper construction, and appliances; and also in providing means for the removal of all air and water arising from condensation; and utilizing

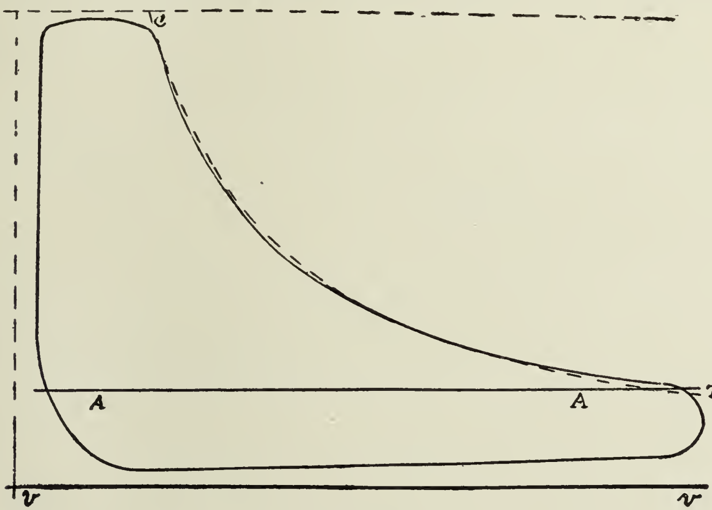


Fig. 68.

such water, as fast as formed by returning it to the boiler; thereby preventing the accumulation of either in the jacket.

In addition to this and to insure the best efficiency it is absolutely essential that the jacket be constantly supplied with dry steam of a temperature fully as high, in all cases, as that of the initial steam entering the steam cylinder; and more especially in engines with early cut-off, and consequently high expansion.

Where the details referred to, are strictly observed and carried out, the result must evidently tend more or less to

better economy and efficiency in engines, by the use of the steam jacket.

On the contrary, if wet steam, or steam containing a large proportion of moisture be introduced into a steam jacketed engine cylinder in the beginning of the stroke, a result will follow, which will be the opposite of economy, and end in considerable loss, this loss arising from the large quantity of heat abstracted from the jacket during the stroke, for the evaporation of this water or moisture that has entered the cylinder.

Therefore to insure that the economy and efficiency which is expected from the use of the steam jacket be realized, it is very essential that all the requirements before mentioned for its proper performance, should be assured, otherwise the jacket may be quite ineffectual; its theoretical efficiency wholly destroyed, and its utility consequently questioned.

In a great majority of cases at the present time, cylinder jacketing is accomplished in accordance with the second condition mentioned; viz, that of thoroughly covering the whole of the exterior surface of the cylinder, and steam chest, with some non-conducting material as felt, wool, or asbestos, and secured thereto by an extra covering of wood or iron; completely enveloping the whole.

This combination of covering where suitably applied, appears to give general satisfaction, as many builders of our best modern engines testify; by the almost exclusive use of some suitable material for cylinder jacketing, on this principle.

Also in many cases where engine cylinders are covered and protected in the manner just described, and having steam tight valves, and piston, it is frequently found that the expansion line of the diagrams therefrom, agree very nearly with the isothermal or true theoretical curve as represented in Fig. 69.

Therefore it is readily seen from the diagrams Fig. 68 and Fig. 69 the advantages to be derived by either of the latter conditions or methods of cylinder jacketing, to secure the greater economy, and efficiency in the engine; *above that* of one from an exposed or unprotected steam cylinder with an

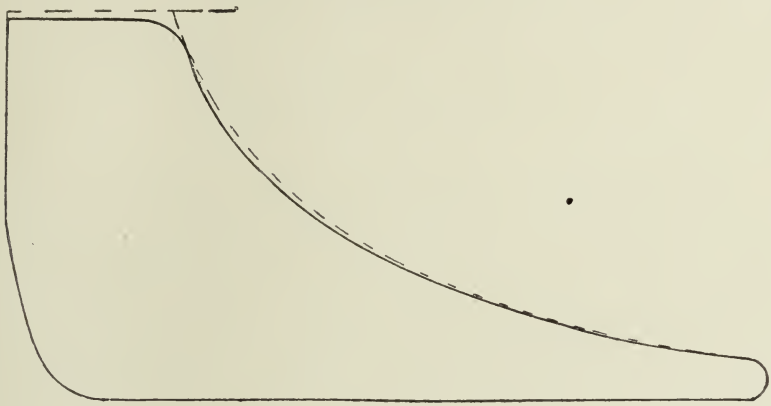


Fig. 69.

early cut-off as represented in the diagram Fig. 67. In order to obtain Indicator diagrams that shall be accurate exponents, and represent the true action of steam expansion within the cylinder, it is absolutely essential that the valves, and piston of the engine be practically steam tight; and also that the Indicator itself have perfect and unimpeded freedom of movement in all its parts, when under pressure and temperature of the steam present. The exact measure of the tension, or in other words, the strength of the spring used, is also of great importance; as the accuracy of all computations based upon the form of the diagram, for obtaining the Mean Effective Pressure, acting against the piston depends upon the correctness of the spring; therefore its accuracy should be determined and established, (by comparison with a correct steam guage), before any elaborate tests are anticipated. This may be accomplished in a satisfactory manner by means of the simple Indicator spring testing device represented and described in Fig. 84, Chapter XXII.

CHAPTER XX.

READING THE DIAGRAM.

The principal and most positive information to be derived from the reading of an actual indicator diagram, is, the measure of the force or pressure in the cylinder, acting upon the opposite sides of the piston, at any and all points, during one complete revolution of the engine; hence the actual card, as compared with the theoretical diagram, (under similar conditions) indicates the efficiency and economy of the engine; and all other information must also be acquired through exceedingly careful consideration, and reasoning in the study of the diagrams; and conclusions arrived at, therefrom in accordance with an exercise of the best judgment of the engineer.

The figures traced by the pencil will vary in outline in different engines, and also from the same engine under varying conditions, due to a number of causes; as leakage of valves, condensation and re-evaporation of the condensed steam in the cylinder, construction and the adjustment of valves, condition of the steam, etc.

These effects will be more apparent along the expansion curve especially; and as a consequence the actual curve will very rarely coincide exactly with the true theoretical curve.

Therefore it is very essential that all cards traced by the indicator should accurately represent the duty performed by the engine; as the accuracy of all such investigations depends entirely upon the correctness of the diagrams.

Upon an examination of the steam expansion curve of indicator diagrams it will be found (almost invariably), that the Terminal pressure is relatively too high (from a given cut-off), as compared with the true theoretical curve; the amount increasing as the ratio of expansion increases: as shown in Fig. 66

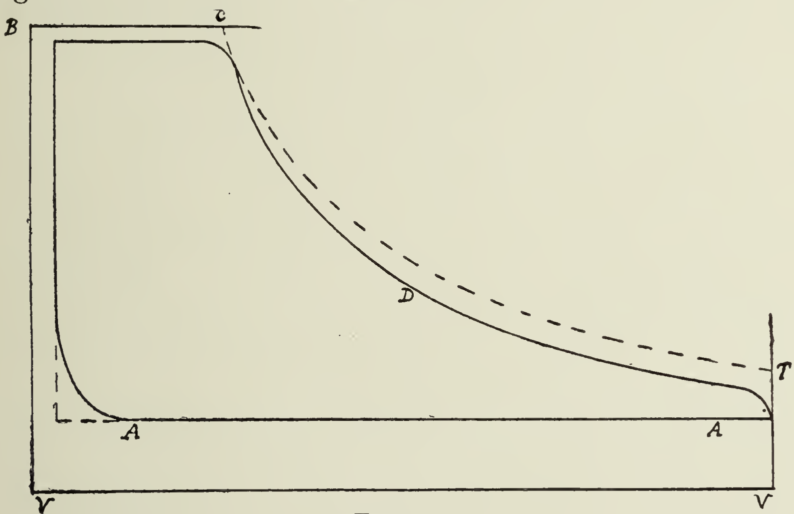


FIG. 70.

This result may be due to either of two causes, or to both combined.

1st. To leaky steam or admission valves, through which the steam is enabled to pass into the cylinder, after the closure of such valve; or in other words, after the point at which cut-off is supposed to have taken place; and thereby producing a higher terminal, than would otherwise appear with steam tight valves.

2nd. To a re-evaporation of the entering steam that is condensed in the earlier part of the stroke, through coming in contact with the interior walls of the cylinder, which have been cooled to a temperature corresponding to the lower pressure of the escaping steam, during its exhaust.

In some cases however the expansion curve of the actual diagram will be found falling below the true or theoretical curve throughout its entire length; as shown in Fig. 70, evidently due, to leaky piston, and exhaust valves; this, also in connection with exposed or unjacketed steam cylinders.

But it frequently happens that cards are found wherein the expansion curve coincides very nearly with the isothermal or theoretical, although they have been taken from engines in which both valves and piston are known to leak badly.

In such cases the leakage through the admission valve *after closing* is just sufficient to restore the steam lost through a leaky piston, and the result under such circumstances, are that the two curves will be a very near approach to each other.

Such cards from observation alone have the appearance of good efficiency and economy in the engine; when in fact the opposite of this prevails, and an extravagant loss and waste of steam is the consequence; all arising from a leaky condition of these parts.

The loss of steam from this cause, and also from cylinder condensation is not accounted for by the indicator, hence does not appear in the diagrams, and is only made apparent by a comparison of the water consumption per horse power (as computed from the diagram), with the actual amount of water that has been supplied to the boiler.

An important matter to be ascertained in reference to all indicators; is, whether the vertical or admission line on the diagram, made by the pencil (when in contact with the paper on the drum) is exactly perpendicular to the atmospheric, or horizontal line, that is made by the pencil, when in contact with the drum while rotating.

A leaning of the admission line either forward or back, thereby causing it to be out of square with horizontal line, tends to be misleading in reference to the proper adjustment of the valves, especially if a fault of this kind exists in the

instrument, and not previously known. This fact may easily be determined at any time before placing the spring in the instrument, by simply placing the paper upon the drum the same as

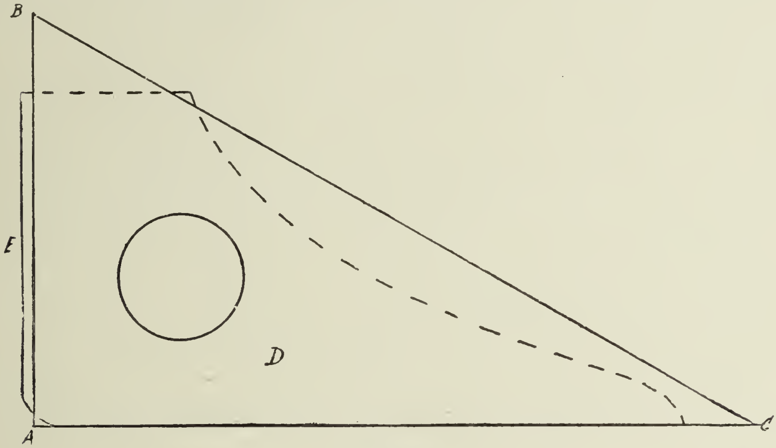


FIG. 71.

for taking diagrams, and bring the pencil in contact with it; then cause the drum to rotate once back and forth by means of the cord attached to the indicator, thereby making a hori-

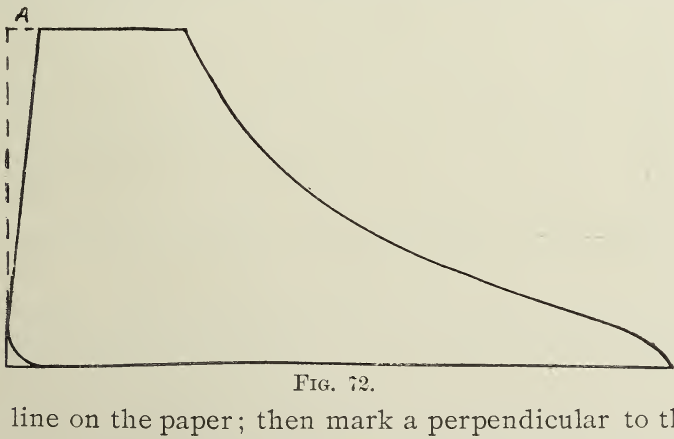


FIG. 72.

zontal line on the paper; then mark a perpendicular to this by raising the pencil by hand to its extreme height. Remove the

paper from the drum, and compare its correctness with any ordinary square or right angle triangle at hand, as shown in Fig. 71. Any inclination or leaning of the admission line in indicator diagrams, either forward or back, as in Fig. 72 and Fig. 73, is usually construed and considered to be an improper adjustment of the valves.

For example. In reviewing the diagram represented in Fig. 72; from the fact of a leaning forward of the admission line, the time of opening of the valve for steam admission, would ordinarily be assumed to be too late, or indicating insufficient lead; and in consequence the piston has advanced a portion of the stroke, (as shown at A), before the initial pressure has reached its highest point; resulting to a certain degree, in a loss of power, and efficiency in the engine.

Again suppose in a diagram the results are as represented in Fig. 73, where the admission line inclines outward; this

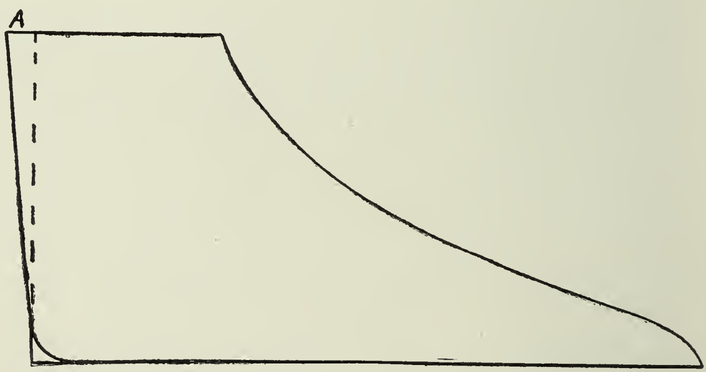


Fig. 73.

might indicate either a too early opening of the steam valve, thereby admitting steam in the cylinder before the piston had completed the stroke; or a too early closing of the exhaust at that end of the cylinder, thus causing excessive compression; either of which would also cause a loss of efficiency and power.

Most types of indicators after their being in use for a length of time, are *liable* to certain defects, or disarrangement more or less in their pencil movement; (and these defects sometimes appears in new instruments) and which prevents the initial pressure or *vertical line* as traced upon the paper, from being at a right angle or perpendicular to the atmospheric line; and in such case the instrument is usually designated as *out of square.*

As a consequence, the admission line at either end of the diagram, will be inclined to the perpendicular, as appears by

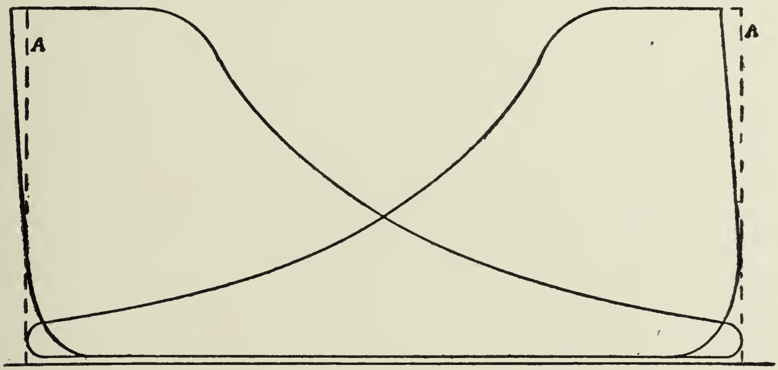


Fig. 74.

the dotted lines A, A, in the diagrams Fig. 74, and which may be wholly due to such incorrectness in the indicator as stated; notwithstanding the valve adjustment of the engine may be practically all that can be desired.

This discrepancy very often arises from careless handling, and from culpable neglect, and abuse of the instrument in various ways. Where a defect of this description exists in the instrument, it is generally advisable to send it at once to the maker, to insure that the necessary correction be properly and satisfactorily accomplished.

CHAPTER XXI.

DIFFERENT METHODS OF COMPUTING THE AMOUNT OF STEAM ACCOUNTED FOR BY THE INDICATOR.

If the number of cubic feet occupied by the steam in an engine cylinder, and the pressure it exerts against the piston at any point of the stroke be known, the number of pounds which the steam weighs may be computed.

The weight of the steam at any point, less the weight remaining in the cylinder at compression, is the weight accounted for by the indicator.

The weight accounted for on one stroke, multiplied by the number of strokes per hour, and divided by the indicated horse power, is the amount of steam accounted for, per indicated horse power per hour.

This computation requires a knowledge of the volume of the clearance space in the cylinder; that is, the cylindrical space between the cylinder head and the piston, when the piston is at the end of its stroke, and also includes the volume of the steam ports and passages which conducts the steam from the admission valve to the cylinder, and also from the cylinder to the exhaust valve.

The water consumption of an engine (as stated in another chapter) is the measure of its economy, but the exact amount is not determinable from the diagram, because that gives the *pressure of steam only*.

Of the water that has been carried over with the steam, or of the steam that has been condensed by coming in contact with the comparatively cold surfaces of the cylinder, the diagram *gives no record.*

We know however, that at least, the amount of steam accounted for by the indicator has passed through the cylinder; and in fact, we know that sometimes a much larger amount has been used.

One method of ascertaining the amount accounted for by an indicator is to calculate the piston displacement, *plus* the clearance space for a given time, in cubic feet, up to a certain point in the stroke before the exhaust opens, and multiply this volume by the weight of a cubic foot of steam of the absolute pressure at this point.

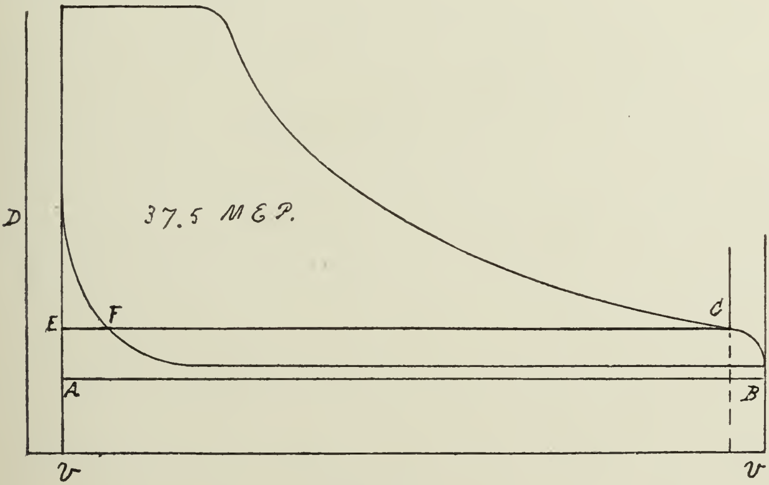


Fig. 75.

This method is described in the diagram Fig. 75 and is as follows: Draw the vacuum line V. V., and the clearance line D. and select some point on the expansion curve, as at C· where it is known that both the steam and exhaust valve are closed and wherever it is possible, have this point C. at a

distance from the end of the diagram equal to the clearance distance D.

Where this can be done, then the volume to point C., including clearance, will just equal the piston displacement which is the area of the piston multiplied by the length of stroke.

Where this cannot be done, then find volume to point C, including clearance, which if multiplied by the weight of a cubic foot of steam of the absolute pressure at the point selected will give the weight of the steam contained in the cylinder at such point.

The volume at point C may be found as follows:

Assume the cylinder of an engine to be 12 inches in diameter, (113.09 square inches in area) with a stroke of piston of 30 inches running 90 revolutions per minute; the area of piston rod being 6.18 inches; the clearance being five per cent. and the scale of the spring 40.

From the area of the cylinder, deduct one-half the area of the piston rod ($6.18 \div 2 = 3.09$ inches) hence; $113.09 - 3.09 = 110$ square inches as the mean area of the piston.

The total piston displacement per stroke therefore will be, the mean area of piston in inches multiplied by length of stroke, also in inches, $110 \times 30 = 3300$ cubic inches per single stroke of the engine.

The displacement to the selected point C, may then be ascertained, by multiplying the total displacement, (3300 cubic inches) by the distance that point C, is from the initial end of the diagram, or from line E, and dividing this product by the total length of the diagram, or line A, B, for example: The length of the line C, E, is 3.56 inches, and the length of A, B, is 3.75 inches, therefore $\frac{3300 \times 3.56}{3\ 75} = 3132.8$, cubic inches.

To this must be added five per cent. for clearance, or $3300 \times .05 = 165$. then $3132.8 + 165 = 3297.8$ cubic inches occupied

by steam at point C. The engine running 90 revolutions per minute consequently makes $90 \times 2 \times 60 = 10800$ single strokes per hour; therefore the displacement per hour is $\frac{3297.8 \times 10800}{1728} = 20611$ cubic feet.

Now with the scale of the spring, (40) with which the diagram was taken, measure the absolute pressure from the vacuum line to point C, and opposite this pressure in the Table No. 8 of properties of saturated steam will be found the weight of a cubic foot of steam at that pressure.

The absolute pressure at point C, is 26 pounds and the weight of a cubic foot of steam at this pressure is .065, hence the weight of steam at this point per hour is $20611 \times .065 = 1339.71$ pounds.

From this amount however the *steam saved* by compression caused by the closing of the exhaust valve before the end of the stroke, *must be deducted;* the remainder being the actual consumption of steam as accounted for by the indicator.

The process by which to ascertain the amount of this deduction is as follows:

From the selected point C, draw a line parallel with the vacuum line to E, intersecting the compression curve at point F.; multiply the accounted consumption by the distance from C to F, and divide this by the distance from C. to E.

The length of the line from C. to F. is 3.32 inches, and that of the line from C. to E. is 3.56 inches, hence $\frac{1339.71 \times 3.32}{3.56} = 1249.39$ pounds of steam exhausted per hour, corrected for both clearance and compression. The Mean Effective Pressure of the above diagram (measured by Planimeter) being 37.5 pounds per square inch, the Horse Power is $\frac{110 \times 37.5 \times 450}{33000} = 56.25$, hence the steam accounted for per horse power per hour will be $\frac{1249.39}{56.25} = 22.21$ pounds.

A somewhat simpler method of determining the same result, is to continue the expansion curve of the diagram in its gradual descent, to the end of the stroke, and by that means locate the terminal pressure as at T, Fig. 76.

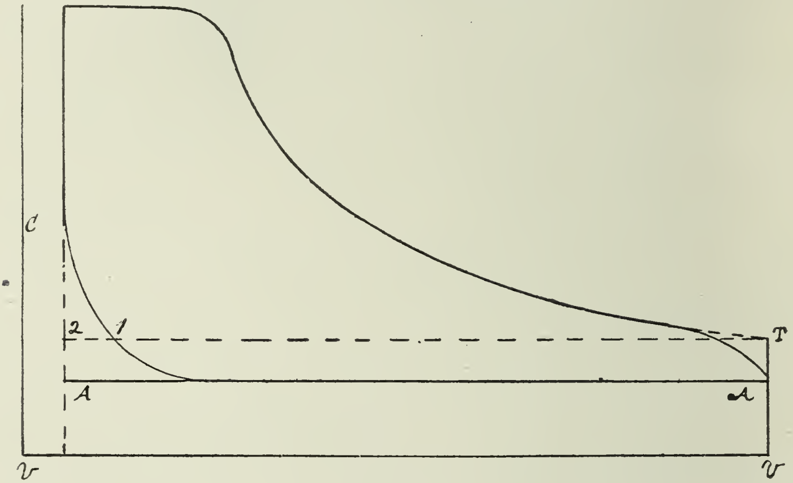


FIG. 76.

This point may be found according to the method described for constructing the theoretical curve as explained in Fig, 56 Chapter XIV, or may be traced by hand from the point of exhaust opening on the expansion curve, to the completion of the stroke; and which will be sufficiently near in cases where no great accuracy is desired.

The process by calculation is to find the total mean displacement of the piston for the *whole* stroke, *plus* the clearance in cubic feet per hour.

Multiply this by the weight of a cubic foot of steam at the terminal pressure T, and divide this product by the Indicated Horse Power.

The quotient is the number of pounds of steam entering the cylinder per horse power per hour. For example: Assuming the engine data to be the same as in the preceeding

example; therefore the mean piston displacement per single stroke in this case will be 3465 cubic inches; which if multiplied by 10800 the number of single strokes per hour, and divide by (1728) the number of cubic inches in a cubic foot, the quotient will be the total piston displacement in cubic feet per hour, that is, $\frac{3465 \times 10800}{1728} = 21656.$

The absolute terminal pressure being 25 pounds per square inch, the weight of a cubic foot of steam at that pressure, according to the table is .063 pound; therefore $21656 \times .063 = 1364.32$ pounds, which divided by the indicated horse power of the diagram (56.25) is equal $1353.50 \div 56.25 = 24.25$ the number of pounds of steam entering the cylinder per horse power per hour.

And this would also be the actual amount of steam exhausted, were it not for the fact that it is exhausted above vacuum from which the pressure is calculated; also a portion of it is *saved* in the clearance space upon the return of the piston, and this *saving* is still further increased by the closure of the exhaust valve before the end of the stroke; therefore the actual indicated consumption will be *minus* this amount.

This correction may be made as follows:

From the terminal pressure T, draw a line T 2, parallel with the vacuum line, thereby intersecting the compression curve at 1, at which point the quantity of steam exhausted from the clearance has been restored, and the consumption will be as much less than the rule shows, as the line T. 1. is shorter than the line T. 2. or the length of the diagram.

Consequently to find the corrected rate, multiply the result as found by the rule (24.25) by the length of the T. 1. = 3.47 inches, and divide by the length of the line T. 2 = 3.75 inches, hence $\frac{24.25 \times 3.47}{3.75} = 22.43$ pounds per Indicated Horse Power per hour; the corrected rate for both clearance and compression.

The volume of the cylinder, and the number of strokes being factors in the computation of the amount of steam accounted for in any given time, and the same also being factors in the calculation of power developed, consequently for this reason it is not necessary to take these quantities into consideration, when only the amount accounted for per horse power per hour is desired.

The above fact provides another, and much more simple method of computing the rate of water consumption; in which the piston displacement is not required, and which is independent of any knowledge of the size or speed of the engine; the diagram alone being sufficient; but it is necessary however to know the mean effective pressure.

This rate may be found by the following rule: Divide the constant number 859375 by the *volume* of steam at the terminal pressure, and by the mean effective pressure.

The quotient will be the water consumption per horse power per hour uncorrected for compression and clearance.

This correction is made in the same manner as that given in the previous method.

This constant 859375 is the number of pounds of water that would be used in one hour by an engine developing one horse power, if run by water, (instead of steam) at one pound pressure per square inch.

The process is based on the following considerations: A standard horse power is 33000 pounds raised one foot per minute, or 33000 foot pounds, which is $33000 \times 60 = 1{,}980{,}000$ foot pounds per hour, or $1{,}980{,}000 \times 12 = 23{,}760{,}000$ inch pounds per hour. The latter number of pounds on being raised one inch per hour, requires the same expenditure of energy, as to lift 33000 pounds one foot per minute; each being the equivalent of the other.

Now suppose the engine to be run by water, (instead of steam) at one pound pressure per square inch, and the number

of cubic inches in a pound of distilled water being 27.648 then 23,760,000÷27,648=859375 which is the desired constant, and which is the number of pounds of water per indicated horse power per hour that would be consumed by an engine driven by water, (instead of steam) at one pound mean effective pressure.

Example: Assume the diagram shown in Fig. 77, to be one from an engine of 12 inches, diameter of cylinder, and 24

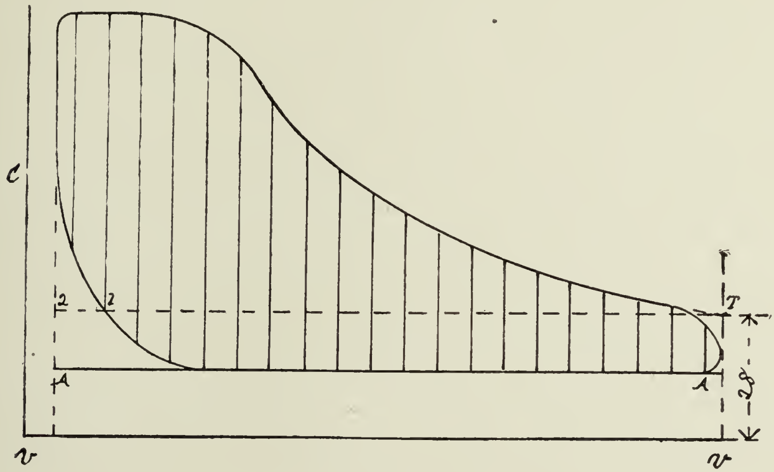


Fig. 77.

inches stroke running 100 revolutions per minute, the scale of the spring 40.

The mean effective pressure of the diagram is found to be 42 pounds per square inch, when measured either by ordinates or by planimeter.

The absolute terminal pressure T. V, is 28 pounds, and the volume at that pressure (as given in table No. 8) is 883, that is one cubic inch of water at a temperature of 60 degrees, makes 883 cubic feet of steam at 28 pounds pressure per square inch.

Hence by the rule the rate of water consumption will be $\frac{859375}{883 \times 42} = 23.17$ of water per indicated horse power per hour.

But in this case some steam is *saved* by the closure of the exhaust valve before the end of the stroke, while some is *wasted* by exhausting from the clearance at a pressure greater than the back pressure, and the above calculation so far makes no allowance for either.

This allowance for compression and clearance may be calculated by the following method:

Locate the point T on the diagram where the expansion line would have terminated, provided the steam had not been released until the end of the stroke.

Draw the line T. 2, parallel with the atmospheric line A. A, which will intersect the compression curve at 1, at which point the quantity of steam exhausted from the clearance has been restored; therefore the consumption will be as much less than the rule shows, as the line T. 1. is shorter than the line T. 2, or the length of the diagram.

Multiply *the result obtained by the rule*, by the length of the line T. 1, and divide the product by the length of the line T. 2; the result will be the rate of consumption corrected for both clearance and compression.

Example: The length of line T. 1, is 3.47 inches, and the length of line T. 2, is 3.75 inches. The rate of consumption obtained by the rule is 23.17, hence $2317 \times 3.47 \div 3.75 =$ 21.43 pounds; *the corrected rate* per indicated horse power per hour.

This latter method is most generally employed by engineers in charge of plants, as it gives a very close approximation, and is very much more convenient than computations made from the steam displacement of the cylinder.

Where diagrams are taken that have only a small amount of compression, the line T. 1, will not intersect the compression

curve; as in Fig. 78. In such cases it is necessary in order to find the length of the line T. 1, to continue the curve from the

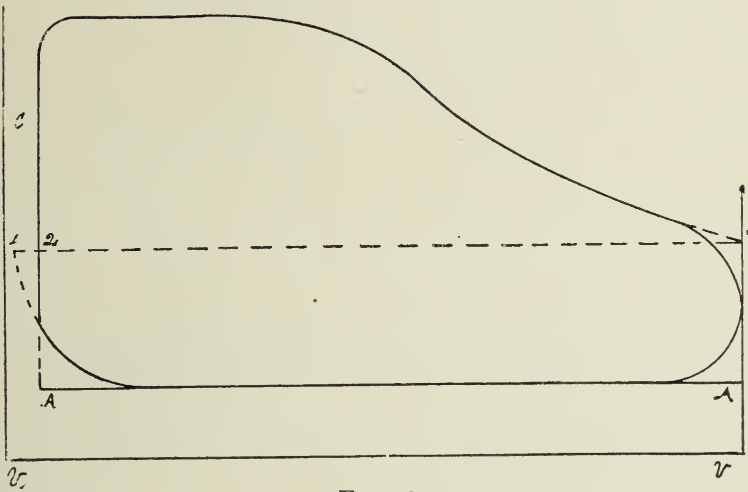


Fig. 78.

end of the diagram, (being guided by the eye) upward in about its natural direction, and far enough beyond the end of the diagram, as to be intersected by the line T. 1, which is always drawn parallel with the atmospheric line, as shown by the dotted lines in Fig. 78.

The line T. 1, will therefore be lengthened, but whatever may be its length, it is *always the multiplier* in making the corrections, while T. 2, is always the divisor, and represents the length of the diagram.

In this case the result obtained by the rule is increased, because the multiplier T. 1, is longer than T. 2.

In diagrams like Fig. 79 where there is no compression, the proper position for point 1 on the terminal pressure line may be found as follows: First draw the vacuum line V, and locate the clearance line C, in accordance with the best data at hand; then draw the terminal pressure line extending from

T. to C, which will intersect the end of the diagram at point 2, and from point 2, draw a diagonal line to the intersection of clearance line with the vacuum line; (see 2 V).

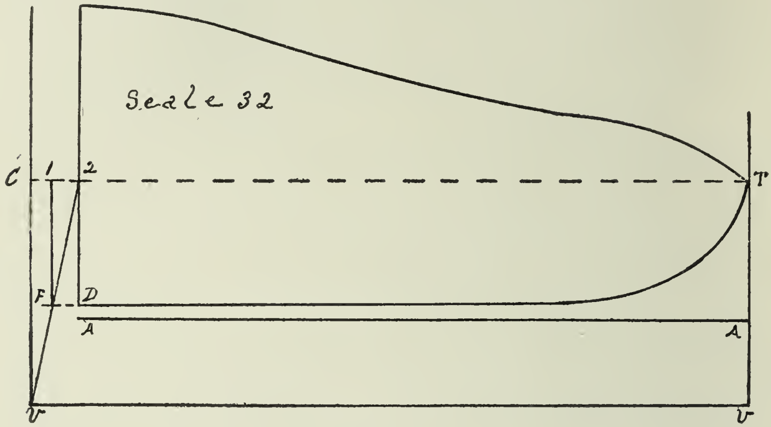


FIG. 79.

This diagonal will intersect a continuation of the back pressure line at F, directly under the proper place for point 1, on the terminal. In this case as in Fig. 78, the result obtained from the rule will be increased, because the multiplier (distance T. 1.) is longer than T. 2 in the correction.

A knowledge of existing clearance is necessary, as such diagrams give no information in regard to it. But the expansion curve of a cut-off diagram however, does furnish the information necessary to arrive at approximately at the volume of clearance, unless the curve is very irregular in its formation. Diagram Fig. 50 illustrates the method of establishing the clearance line by means of the expansion curve.

The diagram Fig. 80 illustrates one process of locating the point 1, (T. 1.) in the terminal line, when this line is below the atmospheric line, and consequently below any part of the compression curve defined on the diagram.

Locate the terminal line by drawing from T, (the terminal pressure) a line parallel with the atmospheric line and intersecting the end of the diagram at 2. Select any point in the compression curve as at D.

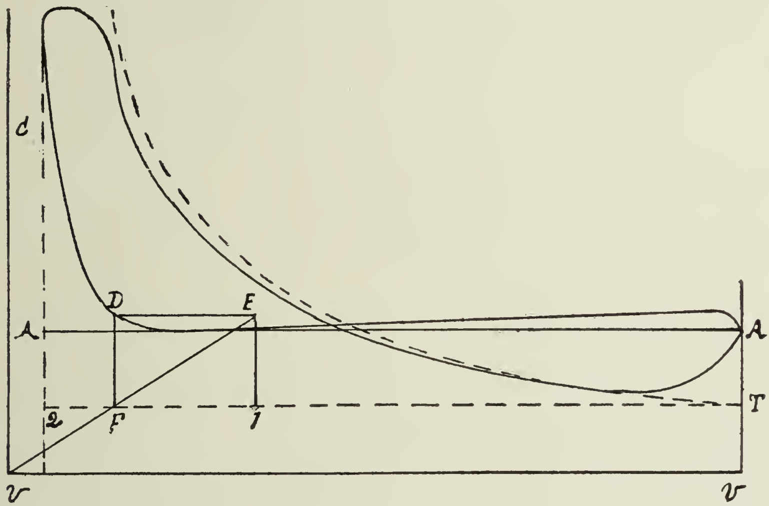


Fig. 80.

From that point draw a line perpendicular to the atmospheric line to terminal line as at F. Then from V where the clearance line intersects the vacuum line, draw a diagonal line through point F, to point E, (same height as point D.) then a line drawn perpendicular to the atmospheric line, from E, will intersect the terminal line at the proper place for point 1.

The process will be recognized the same in principle as that used for finding a point in the isothermal expansion curve.

The water consumption computed for diagram Fig. 80 is as follows:

The mean effective pressure as measured by planimeter is 2⅛ pounds per square inch, and the terminal pressure is 7 pounds absolute.

The volume for 7 pounds as given in the Table No. 8, is 3300, hence $\frac{859375}{3300 \times 2.125} = 122.5$ pounds uncorrected for compression.

Line T. 1, is 2.60 inches long, and line T. 2C(or the whole length of diagram) is 3.75 inches, hence $122.5 \times 2.60 \div 3.75 =$ 84.93 pounds per indicator horse power per hour, the correct rate.

Diagrams similar to Fig. 80 are wasteful of steam, and are usually obtained from engines having insufficient load, and a comparison of the water consumption, (as computed) with that of diagrams taken from moderately loaded engines will at once make apparent the economy of the latter, as against the extravagant waste of the former.

Probably no other single condition is so detrimental to good economy as an engine over large for its work, as a too light load necessitates an early cut off; the expansion and consequent fall of temperature becomes excessive, and hence internal condensation appears to the fullest extent.

In the computation for water consumption of these diagrams it must be understood that the rates as calculated are theoretical, and assumes perfect conditions, such as dry steam, entire absence of loss from leakage, condensation, etc.

The diagram shows *only* the minimum amount of steam that has been consumed by the engine to do a given amount of work, and there are *many reasons why* this consumption of water as shown by the indicator should *be less* than the actual amount.

It is considered that the percentage of loss in a modern and properly constructed steam plant is fully twenty per cent.; and taking the engine alone, is at least ten per cent. and this may be still further increased by condensation, and also where considerable leakage occurs, etc., so that it is safe to add at least 10 per cent. to the indicated consumption to closely approximate the actual consumption.

The loss from water that is carried over with the steam is chargeable to the boiler, and not to the engine.

The unindicated loss will also be greatest at light loads. With steam at 80 pounds pressure, and a mean effective pressure of from about 40 to 45 pounds, (corresponding to about one-fourth cut-off), will give the least loss.

Very short cut-off gives an increased loss.

The steam used in the low pressure cylinders of compound engines, first passes through the high pressure cylinder; hence the water consumption as computed for the high pressure cylinder, (corrected for compression) will be the measure of consumption for the whole engine.

This amount is to be divided by the horse power of the whole engine for the consumption per indicated horse power per hour.

The consumption may also be computed from the low pressure cylinders in the same manner as for the high pressure cylinder; but which will be found to disagree with the former owing to some loss between the cylinders.

It will also be found that if no other steam is admitted to the low pressure cylinders, except what has already passed through the high pressure cylinder, that the water consumption will appear greatest when calculated from the high pressure, and will gradually become less from each successive cylinder: therefore it is a good plan, and of interest to measure the consumption from each and all of the cylinders, and compare the results.

The differences may be considered as fair measures of the loss in transmission between the cylinders.

Another method is here given for calculating the water consumption *by constant*, and which is also independent of any knowledge, (except the mean effective pressure) of the size or speed of the engine; and which may be easily and accurately determined from the diagram, for both cut-off, and release by

means of, and the use of the formula: $\frac{13750}{\text{M. E. P.}} \times$ (proportional volume at *cut-off* $\times$ weight of steam) minus, (proportional volume at compression $\times$ weight of steam) = number of pounds of steam accounted for at *cut-off*, per indicated horse power per hour.

Or, by the formula: $\frac{13750}{\text{M. E. P.}} \times$ (proportional volume at release $\times$ by weight of steam) minus, (proportional volume at compression $\times$ weight of steam) = number of pounds of steam accounted for at *release* per indicated horse power per hour.

The following is the explanation of the above formula:

M. E. P. is the mean effective pressure. In compound, triple and quadruple expansion engines, this is the sum of two or more quantities.

One is the M. E. P. of the cylinder under consideration, as for instance, the high pressure cylinder, and the others are the M. E. P. in the other cylinders referred to the high pressure cylinder.

The proportional volume at cut-off is the percentage of the stroke computed at cut-off, (as at D. Fig. 81) added to the percentage of clearance, and this is to be multiplied by the weight of one cubic foot of steam at the cut-off pressure.

The proportional volume at compression is the percentage of the return stroke uncompleted at compression added to the percentage of clearance, and this is to be multiplied by the weight of one cubic foot of steam at the pressure where compression begins.

The proportional volume at release is the percentage of the stroke completed at release added to the percentage of clearance, and this is to be multiplied by the weight of one cubic foot of steam at the pressure where release is taken.

The *constant* 13750 is the volume of steam in cubic feet per hour required by an engine without clearance to develop

one horse power when working with one pound pressure per square inch and without expansion. This quantity will be less in proportion to the increase of average pressure; therefore it is divided by the mean effective pressure (M. E. P.) of the diagram.

The quantity will also be increased in proportion to the percentage of clearance, and decreased by the quantity of steam saved by compression.

The points of cut-off, release, and compression referred to are shown respectively at D. E. and F. in the diagram Fig. 81.

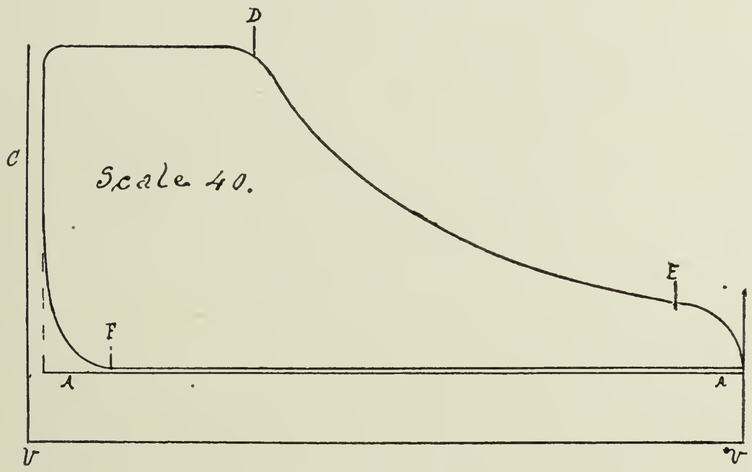


Fig. 81.

The pressures at these points must be the absolute pressure, taken from zero or a perfect vacuum, which is 14.7 pounds below the atmosphere when the barometer indicates 29.92 inches.

Where great accuracy is desired, the height of the barometer should be observed, when the diagrams are taken in order that the atmospheric pressure which it shows should be used in such cases.

The pressure of the atmosphere as shown by the barometer in inches of mercury should be multiplied by 0.491 to reduce it to pounds per square inch.

The requisite data for computing the amount of steam accounted for, from the diagram Fig. 81 is as follows: The proportion of stroke completed at cut-off D, is thirty-hundredths, (.30) and the absolute pressure of steam per square inch at that point is eighty-three (83) pounds.

The weight of a cubic foot of such steam is .1967 pounds.

The proportion of stroke completed at release E, is ninety-hundredths (.90) and the absolute pressure thirty (30) pounds.

The weight of a cubic foot of such steam is .0755 pounds.

The proportion of the return stroke uncompleted at compression F. is ten-hundredths (.10) the absolute pressure is sixteen (16) pounds, and the weight of a cubic foot of this steam is .0413 pounds.

The clearance of the engine equals .02 per cent.

The mean effective pressure on the piston is forty (40) pounds per square inch. Hence by this method the amount of steam accounted for at *cut-off* will be $\frac{13750}{40} \times (.30+.02).1967 - (.10+.02).0413 = 343.75 \times .0579 = 19.90$ pounds of steam or water per indicated horse power per hour.

The amount of steam accounted for at *release* will be: $\frac{13750}{40} \times (.90+.02).0755 - (.10+.02).0413 = 343.75 \times .0645 = 22.17$ pounds of steam per hour.

Suppose in the engine to which these calculations apply, an actual feed water test gave a consumption of thirty (30) pounds of water per indicated horse power per hour; then the percentage of feed water accounted for at *cut-off* is $\frac{19.90}{30} = .663$ and at *release* $\frac{22.17}{30} = .739$.

The formula for release may be simplified in cases where there is a sufficient amount of compression, by locating the compression point at such a height on the compression curve as will make that pressure and the release pressure equal, as shown in diagram Fig. 82.

The formula then becomes $\frac{13750}{\text{M. E. P.}}$ (percentage of stroke completed at release, minus percentage uncompleted at com pression) multiplied by the weight of a cubic foot of steam at release pressure, thus, $\frac{13750}{\text{M. E. P.}}(.90-.10).0755=20.62$ pounds, the effect of clearance disappearing. The quantity in the parenthesis is the proportion which the distance (1) between the points in Fig. 82 bears to the whole length of the diagram

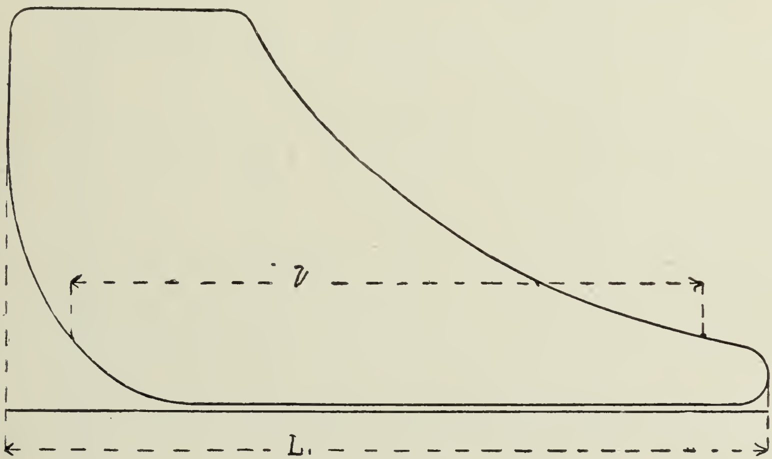


Fig. 82.

or to L. That is the proportion $\frac{1}{L}$ This applies only to the release formula.

The calculation of the quantity $\frac{13750}{\text{M. E. P.}}$ in these formulas may be facilitated by reference to following Table No. 5,

QUANTITY OF STEAM ACCOUNTED FOR BY INDICATOR.

M.E.P. lbs.	13750 / M.E.P.	M.E.P. lbs.	13750 / M.E.P.	M.E.P. lbs.	13750 / M.E.P.	M.E.P. lbs.	13750 / M.E.P.
10.	1375.0	36.5	376.8	66.	208.3	119.	115.5
10.5	1309.6	37.	371.6	67.	205.2	120.	114.5
11.	1250.0	37.5	366.7	68.	202.2	121.	113.6
11.5	1195.6	38.	361.9	69.	199.3	122.	112.7
12.	1145.8	38.5	357.2	70.	196.4	123	111.7
12.5	1100.0	39	352.6	71.	193.7	124.	110.8
13.	1057.7	39.5	348.2	72.	191.0	125.	110.0
13.5	1008.6	40.	343.8	73	188.4	126.	109.1
14.	982.1	40.5	339.6	74.	185.8	127.	108.2
14.5	948.2	41.	335.4	75.	183.3	128.	107.4
15.	916.7	41.5	331.4	76.	180.9	129.	106.5
15.5	887.0	42.	327.4	77.	178.6	130.	105.7
16.	859.4	42.5	323.6	78.	176.3	131.	104.9
16.5	833.4	43.	319.8	79.	174.1	132.	104.1
17.	808.8	43.5	316.0	80.	171.9	133.	103.3
17.5	785.8	44.	312.6	81.	169.8	134	102.6
18.	763.9	44.5	309.0	82.	167.7	135.	101.8
18.5	743.2	45.	305.6	83.	165.7	136.	101.1
19.	723.7	45.5	302.2	84.	163.7	137.	100.3
19.5	705.2	46.	298.9	85.	161.8	138.	99.6
20.	687.5	46.5	295.6	86.	159.9	139.	98.9
20.5	670.8	47.	292.6	87.	158.0	140.	98.2
21.	654.8	47.5	289.4	88.	156.2	141.	97.5
21.5	639.6	48.	286.5	89.	154.5	142.	96.8
22.	625.0	48.5	283.6	90.	152.8	143.	96.1
22.5	611.2	49	280.6	91.	151.1	144.	95.4
23.	597.8	49.5	277.8	92.	149.4	145.	94.8
23.5	585.2	50.	275.0	93.	147.8	146.	94.1
24.	572.9	50.5	272.3	94.	146.3	147.	93.5
24.5	561.2	51	269.6	95.	144.7	148.	92.9
25.	550.0	51.5	267.0	96.	143.2	149.	92.2
25.5	539.2	52.	264.4	97.	141.8	150.	91.6
26.	528.9	52.5	261.9	98.	140.3	151.	91.0
26.5	518.8	53	259.4	99.	138.9	152.	90.4
27.	509.3	53.5	257.0	100.	137.5	153.	89.8
27.5	500.0	54.	254.6	101.	136.1	154	89.2
28.	491.1	54.5	252.3	102.	134.8	155.	88.7
28.5	482.4	55	250.0	103.	133.4	156.	88.1
29.	474.1	55.5	247.7	104.	132.2	157.	87.5
29.5	466.2	56.	245.5	105.	130.9	158.	87.0
30	458.3	56.5	242.4	106.	129.7	159.	86.4
30.5	450.8	57.	241.2	107.	128.5	160.	85.9
31.	443.5	57.5	239.1	108.	127.3	161.	85.4
31.5	436.6	58	237.1	109.	126.1	162.	84.8
32.	429.7	58.5	235.1	110.	125.0	163.	84.3
32.5	423.0	59	233.1	111.	123.8	164.	83.8
33.	416.7	59.5	231.1	112.	122.7	165.	83.3
33.5	410.4	60.	229.2	113.	121.6	166.	82.8
34.	404.5	61.	225.4	114.	120.6	167.	82.3
34.5	398.6	62.	221.7	115	119.5	168.	81.8
35.	392.9	63.	218.3	116.	118.5	169.	81.3
35.5	387.4	64	214.9	117.	117.5	170.	80.8
36.	381.9	65.	211.5	118.	116.5	171	80.4

QUANTITY OF STEAM ACCOUNTED FOR BY INDICATOR.—Continued.

M.E.P. lbs.	$\frac{13750}{\text{M.E.P.}}$	M.E.P. lbs.	$\frac{13750}{\text{M.E.P.}}$	M.E.P. lbs.	$\frac{13750}{\text{M.E.P.}}$	M.E.P. lbs.	$\frac{13750}{\text{M.E.P.}}$
172.	79.9	193.	71.2	213.	64.5	233.	59.0
173.	79.4	194.	70.8	214.	64.2	234.	58.7
174.	79.0	195.	70.5	215.	63.9	235.	58.5
175.	78.5	196.	70.1	216.	63.6	236.	58.2
176.	78.1	197.	69.7	217.	63.3	237.	58.0
177.	77.6	198.	69.4	218.	63.0	238.	57.7
178.	77.2	199.	69.0	219.	62.7	239.	57.5
179.	76.8	200.	68.7	220.	62.5	240	57.2
180.	76.3	201.	68.4	221.	62.2	241.	57.0
181.	75.9	202.	68.0	222.	61.9	242.	56.8
182.	75.5	203.	67.7	223	61.6	243.	56.5
183.	75.1	204	67.4	224.	61.3	244.	56.3
184.	74.7	205.	67.0	225.	61.1	245.	56.1
185.	74.3	206.	66.7	226.	60.8	246.	55.8
186.	73.9	207.	66.4	227	60.5	247.	55.6
187.	73.5	208	66.1	228.	60.3	248.	55.4
188.	73.1	209.	65.7	229.	60.0	249.	55.2
189.	72.7	210.	65.4	230.	59.7	250	55.0
190.	72.3	211.	65.1	231.	59.5	251.	54.7
191.	71.9	212.	64.8	232.	59.2	252.	54.5
192.	71.6						

The foregoing table gives the result of the division for each half pound mean effective pressure, between 10 and 60, and for each pound between 60 and 252.

It is a good plan to compute the steam accounted for, at both cut-off and the release points of the diagram; because if the expansion curve should deviate much from the isothermal a very different result is shown at one point from that shown at the other.

In many cases the extent of the loss occasioned by cylinder condensation and leakage is indicated in a more truthful manner at the cut-off than at release.

The constant 13750 may also be employed in a somewhat similar manner for computing the steam consumption of an engine, by the following method.

Select any point as at D. on the expansion curve, Fig. 83, and draw a line from it, and parallel with the vacuum line,

until it intersects the compression line at C, then with the scale of the spring, (measuring from vacuum to the height of this line above) find the pressure of steam at such height; then from Table No. 8, find the weight of a cubic foot of steam at that pressure.

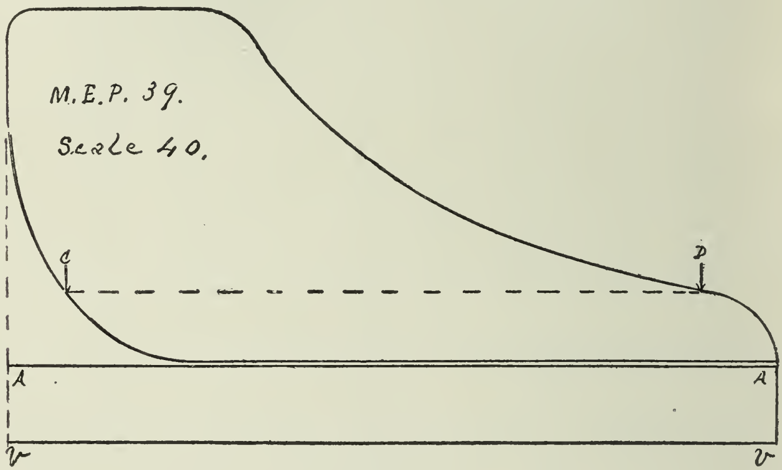


Fig. 83.

Multiply 13750 by this weight of steam per cubic foot, and by the distance in inches, between the points C. and D. Divide the product by the mean effective pressure multiplied by the distance A. A. or the extreme length of the diagram.

The result will be the number of pounds of steam consumed per indicated horse power per hour as shown by the diagram.

For example: Suppose the whole length of the diagram to be 3.75 inches, and the distance between the points C. and D, 3.10 inches, the scale of the spring 40, and the mean effective pressure 39 pounds per square inch, the pressure from vacuum to line C. D. being 30 pounds. The weight of a cubic foot of this steam as shown by the table No. 8 is .0755 pounds.

Therefore $\frac{13750 \times .0755 \times 3.10}{39 \times 3.75} = 22.$ pounds consumption per indicated horse power per hour.

The use of the constant number 13750 is based on the following considerations:

If a piston one square inch in area moves twelve inches, it will do work equal to *one foot pound* for each pound pressure of steam per square inch. That is, every twelve cubic inches of piston displacement represents one foot pound of work at one pound mean effective pressure; and as twelve cubic inches is equal to $\frac{1}{144}$ of a cubic foot, the piston must sweep a volume of $\frac{33000 \times 60}{144} = 13750$ cubic feet per horse power per hour, when the mean pressure equals unity; therefore as the volume of steam used per horse power per hour varies inversely as the mean effective pressure and if the weight of a cubic foot of steam at the release pressure be designated by W, and the mean effective pressure by M. E. P. we have the formula, $\frac{13750}{\text{M. E. P.}} \times \text{W} =$ the number of pounds of water consumed per indicated horse power per hour, exclusive of waste by condensation and leakage; and also makes no allowance for clearance and compression.

CHAPTER XXII.

INDICATOR TESTING DEVICE.

In indicator practice it is frequently found that the initial pressure in the steam cylinder, as shown by the indicator diagram, will in some cases be from five to ten or twelve pounds less pressure per square inch, than the pressure in the boiler, as indicated by the steam guage.

This discrepancy in pressure between the indicator and steam guage may arise from various causes; such as inadequate size of steam pipe, also tortuous and rough passages, non-covered or unprotected pipes, incorrect valve setting, tardy valve motion, etc.; all or any of which tend to cause an apparent difference of pressure between the indicator and steam gauge.

Where such differences do exist in these pressures, the fault is generally supposed at first thought, to lie with the indicator, when in fact it may be due to any of the causes named; or may be due to the incorrectness of the steam gauge itself.

Therefore it becomes important that means be taken to ascertain how near the gauge and indicator agree in denoting the steam pressure, in order that the amount of pressure lost between the boiler and engine may be determined.

For making such comparative tests the arrangement illustrated in Fig. 84, is easily constructed, not very expensive, and is well adapted for the purpose.

It may be connected directly with the steam space of the boiler, or may be attached to the steam pipe in any convenient position.

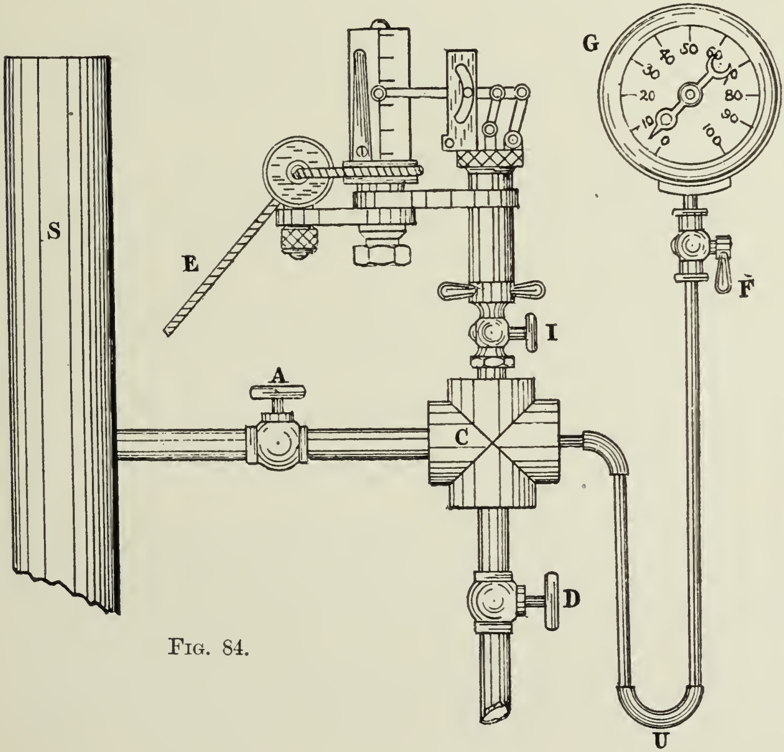


FIG. 84.

In making a comparative test when the device is attached to the steam pipe (as shown) it is best that it be done at a time when the engine is at rest, and the throttle valve closed in order to avoid any fluctuations of pressure that otherwise might exist in the pipe.

In the matter of construction the Chamber C. should be of such size as to contain a considerable volume of steam, and such chamber may consist of an ordinary four inch cross fitting, with the inlet and outlets reduced to suit the size of pipe employed.

Steam is admitted to the system through a 1 inch pipe, and controlled by the globe valve A, and discharged as occasion requires, through the 1¼ inch pipe by the valve D. This pipe is made larger in order to facilitate the discharge of the volume of steam under pressure in the chamber, as quickly as possible.

The top is tapped to suit the indicator cock I, which is usually made of a size corresponding to ½ inch pipe fittings.

The bent pipe U, leading to the steam guage G, may be of ¼ inch pipe.

The bend in the guage pipe should be extended downward at least four or five feet below the center of the four inch cross fitting; thereby securing a sufficient column of water to insure against overheating of the gauge above the existing medium, or atmosphere.

Before preparations for any tests are made, it will be advisable to partly fill the gauge pipe with water; which can be readily accomplished by disconnecting the gauge G, and pouring water into the pipe until it stands in both legs of the pipe about as high as the center of the chamber C.

In preparing to make a test, attach the indicator containing its spring, and also steam guage, in the manner shown in the illustration, and secure a piece of paper in the usual way to the drum of the indicator. Close the discharge valve D, and open the indicator cock I, also open communication between indicator, and gauge, by means of cock F.

Now by gradually admitting steam to the chamber by the valve A, there will be a simultaneous advance movement of both the indicator pencil, and gauge hand, and which will

continue as the steam is admitted until the desired limit of the indicator spring has been reached. This is a preliminary operation for the purpose of warming up the indicator preparatory to making the card.

After the indicator and its spring has been thoroughly warmed, first close the valve A, and then open the valve D, thus discharging the steam from the chamber, through the open pipe into the atmosphere; and thus lowering the pressure, and causing the indicator pencil and gauge hand to return each to their normal positions.

Now with one hand, bring the indicator pencil in contact with the paper on the drum; and by means of the cord E, (with the other hand) cause the drum to rotate a small amount, which in consequence results in tracing a line upon the paper, and which represents the zero or atmospheric line.

Then close the discharge valve D, and slowly admit steam to the chamber through the valve A, and continue until the observed reading of the steam gauge denotes, say ten pounds pressure per square inch.

Just at this point mark another line on the paper by the same means, (by hand) as employed in tracing the atmospheric line.

Continue to mark the corresponding lines on the paper for each successive ten pounds movement of the gauge hand (from observation) until the pressure limit of the indicator spring has been reached.

Close the admission valve A, and discharge the steam remaining in the chamber through the valve D.

Remove the paper from the drum, and compare the marking with a rule or scale, on which the divisions coincide with pounds pressure per square inch, according to the denomination of the indicator spring used.

Supposing a forty pound spring is to be used in the indicator, and assuming both the steam gauge and spring as

correct; then the marking would appear as shown at A in the illustration Fig. 85 for each ten pounds on the gauge, successively from zero to eighty pounds pressure per square inch. But in many cases they do not agree so uniformly, as shown in the figure from the fact than steam gauges, and also indicator springs vary more at some pressures, than at others; hence, such a test enables the operator to observe the true action of the spring; also at what part of the marking the greatest variations (if any) occur, and shows that some springs although correct in some parts of their compression, are incorrect in other parts; and also that either gauge or spring may show light at some pressure and heavy at another.

If the spring registers the greater pressure according to its scale it is light, and if less it is heavy, provided the steam gauge is correct.

This device is easily manipulated to mark a *descending*, as well as an ascending pressure, and on the same paper, and may be accomplished by a very gradual releasing of the pressure in the chamber, after the extreme height (to which the indicator spring should be subjected) has been reached, and in again marking the paper, upon the descent of the pencil, from the same readings of the gauge as was done in the ascending pressure.

A comparison of this kind is both interesting and instructive, as it furnishes the means for observing the various phenomina connected with springs in general, and in their application to different purposes.

In a test of this description, and where friction exists in the indicator, it is found that a variation or lack of coincidence more or less, appears in the lines so marked; that is, a difference between the lines marked when the pencil is rising and those marked when the pencil is under falling pressure; the latter failing (particularly at the higher pressures) to drop sufficiently low, as to meet the lines marked during a rising

pressure: A corresponding pressure always being denoted by the steam gauge at each marking up or down.

This lack of coincidence, gradually decreases from the higher pressures downward until the zero line has been reached, and where the lines again agree.

This is shown at B, Fig. 85, and the column marked *up*, is the rising, and that marked *down* the falling pressure.

The fact of their disagreement is caused principally to *undue friction* in some part of the indicator, and *might* also be

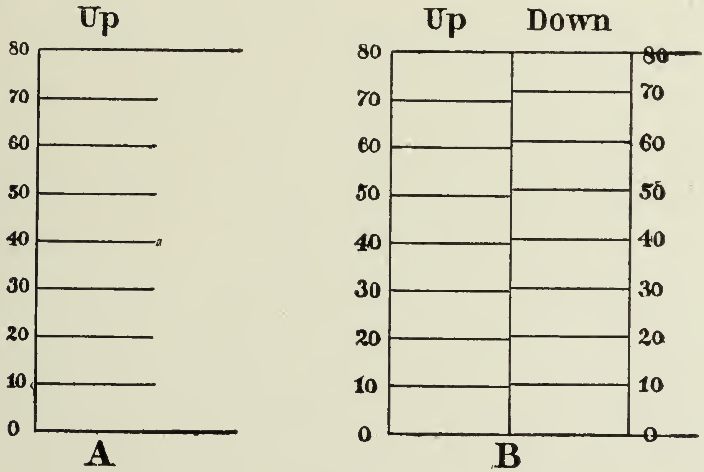


Fig. 85.

partly due under some circumstances to lost motion in the pencil mechanism.

Consequently the elimination of *friction* in the indicator to a minimum, is a matter greatly to be desired; it being an important requirement in all indicators, in order to insure accuracy in the diagrams.

Although an indicator, upon inspection may appear satisfactory in all respects *when cold*, it may become the reverse of this when in operation and subjected to a high temperature of steam; this, application of heat, and circumstances causing

unequal expansion of the metals of which the indicator cylinder is composed.

The expansion that takes place upon being heated, varies in the different parts of the indicator; generally increasing the size of the piston to a greater extent than it does the surrounding metal; and thereby involves a liability of the indicator piston to become sufficiently increased in size as to bind in the cylinder; thus creating excessive friction, but which may obviously be eliminated by a slight reduction in the size of the piston.

Another source of friction which often happens, arises from springs of imperfect construction, or out of true, causing when under tension, a lateral or side pressure against the cylinder.

Either of these faults results in an interrupted or broken action in the movement of the pencil, and which is fatal to the accuracy of the instrument, therefore in order that perfect freedom of action, and that smoothness and accuracy in the pencil movement be attained, it is indispensable that friction in the instrument be reduced to the lowest degree possible.

By the use of this device an amount of interesting and varied information may be obtained, pertaining to the condition and action of springs, the variations in pressure for equal movement of the pencil, in showing the difference between a rising and falling pressure when undue friction is present, and also as a means of observing inaccuracies that may appear in any part of the mechanism connected with the indicator.

In many cases errors arise from excessive friction of the indicator piston, caused by scale or grit of any description being carried from the pipes and other connections leading to it.

If such should be suspected it will likely be detected (where slow speeds prevail) either by close observation of the pencil in its movement up and down, or by placing the finger

at the top of the indicator piston rod, and gently follow it in its downward movement.

As a matter of course the remedy is to remove the piston and clean.

Sometimes with new indicators and clean pistons an unusual amount of piston friction shows itself in the diagram by a series of very definite serrations on the expansion line just after cut-off, as shown in Fig. 86 the horizontal portion of the serrations indicating a disposition of the piston to hang at each of these positions in its descent.

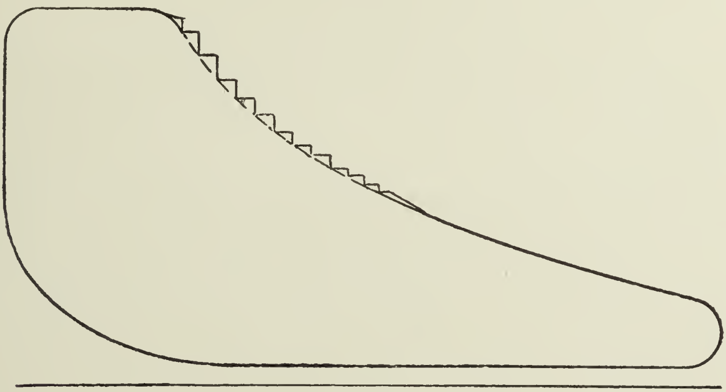


FIG. 86.

In some cases of this kind it may be necessary to very slightly reduce the size of the piston by means of a fine crocus paper, or by oiling, and allowing it to run a short time, having first disconnected the pencil movement.

With springs of a higher tension, that is, with stronger springs, the serrations resulting from the friction of a tightly fitting piston, will not be so apparent, and will be less defined, than with the lighter springs. However the difference in results on the diagram between a tight piston, and one fitting freely, will be, that with the former the various events of the stroke such as cut-off, release, and compression, will occur

later in the stroke, owing to the tardy response of the piston to the variations of steam pressure.

The fricton of a tightly fitting piston therefore will cause the initial pressure to be *less*, but the pressure along the expansion, and also the back pressure line, (on account of its tardiness) will be *greater*, than with the more freely moving piston.

However the area of diagrams from each may not differ greatly; because the loss of initial pressure is partially compensated in the formation of the expansion line; owing to a tardy piston.

Occasionally the mean effective pressure in each may not differ materially, still in most diagrams that have been taken with an indicator in which the piston was too tightly fitted, the diagrams have been found to be unreliable, inaccurate, and misleading.

The piston friction on an indicator may be approximately determined in the following manner: First allow the instrument (by a few working strokes) to become heated to a temperature coinciding with the steam pressure present; then after closing communication with the engine cylinder, gently depress the pencil lever (by hand) just sufficient to slightly extend the spring, and then allow it to slowly return to rest.

While in this position a horizontal line is drawn on the diagram.

The pencil lever is next raised and the spring slightly compressed, and then again allowed to come to rest and another line drawn as before.

The distance or space between the lines so marked is a measure of the sum of the total frictional resistance in both directions, and assuming the pencil movement without friction, then the whole of the error so measured is attributable to piston friction.

Careful attention to the lubrication of the piston, and pencil movements will conduce to smooth running, and to a certain

extent, will prevent the tendency to stick or bind in the cylinder.

Clean cylinder oil will be found a far superior lubricant for the piston, than the limpid oil used for the pencil movements.

CHAPTER XXIII.

PLANIMETERS.

Where considerable care and attention is used the mean effective pressure of diagrams may be computed with a close approximation to accuracy, by the use and method of ordinates, as before described, but the operation is usually attended with considerable anxiety, and also becomes otherwise a rather tedious operation in various ways with more or less liability of error. Therefore, where the mean effective pressure of a large number of diagrams is desired, and where greater accuracy is required, time and labor may be greatly facilitated by the use of an instrument termed a Planimeter, constructed and used for the purpose of correctly measuring the area of any irregular figure regardless of its outline. There are various forms of this instrument, some of which are so constructed that by moving the tracing point over the entire outline of a diagram, its area may be read from a graduated index wheel, its movement being relative to some fixed or zero point. There are others in which the reading is taken from the movement of a blank wheel, traversing a graduated scale, and the result is ascertained by noting the coincidence on the scale of some particular line corresponding to the edge of the wheel that has been selected, and made to coincide with a zero point of the scale; when commencing to trace the diagram.

The instrument shown and illustrated in Fig. 87 is one of the former, with a graduated index wheel and vernier and

represents the well known Coffin Averaging Planimeter in position on its board; and which was especially designed and adapted to the purpose of measuring the mean effective pressure of indicator diagrams. With this instrument no calculations whatever are required to ascertain the average pressure or mean height of the diagram throughout the stroke of the engine, and it may also be applied, when desired, for measuring the areas of any, and all other irregular figures. It is especially valuable where a large number of diagrams have to be measured for area or mean effective pressure, either or both of which may be ascertained at all times (without any adjustment of the instrument), by a single passage of the tracing point around the outline of the figure. In consideration of the accuracy attained and the ease with which it is manipulated makes its use desirable in all cases in a single as well as in a number of diagrams, as the chances of error in making calculations are entirely eliminated.

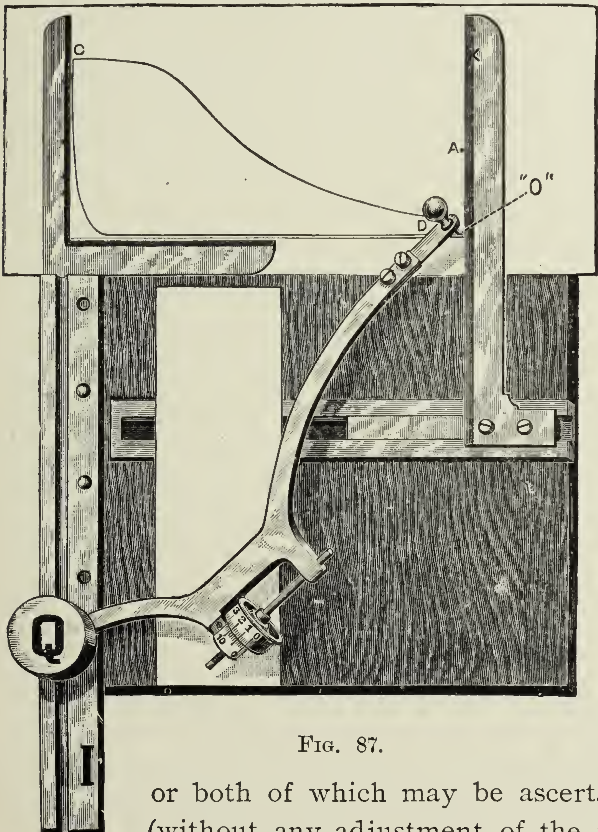


Fig. 87.

The parts of this instrument being permanently secured insures it always ready for use, without the necessary adjustment to length of diagram, etc., required by some other makes, in order to ascertain areas and average pressures, and which gives only the readings of one or the other separately. This instrument measures the area and average pressure or mean height of any diagram or figure at the same operation, however irregular, or whatever its shape may be, just as quickly and accurately as if it were some regular figure, such as a square or rectangle.

The Coffin Averaging Instrument proper consists of an arm fitted at one end with a tracing point O, and at the other with a hardened steel guide pin (not shown in the cut), the centre of which is common with the centre of the weight Q. Upon this arm is also mounted a graduated index wheel and spindle, delicately poised on hardened steel centres, thereby reducing friction to a minimum. The axis of said wheel being parallel to a line drawn from the centre of a guide pin to the tracing point. In close proximity to the wheel there is permanently secured to the arm a graduated vernier scale and used in connection with the graduations of the wheel, thereby enabling the readings to be readily observed in small fractional parts of a square inch. The distance between the centres of the guide pin and the tracing point of the arm is assumed to represent the length of one side of a rectangle, while the circumference of the graduated wheel represents its component or height, and if the terms of these two factors be in inches, the product of their multiplication will be the area of the rectangle of such dimensions in square inches. For example: In this instrument the distance between the guide pin and tracing point is six and one-quarter (6.25) inches, and the circumference of the wheel two and four-tenths (2.4). corresponding to a diameter of about .764 part of an inch, then 6.25x2.4 is equal to fifteen square inches, which will be the area of a

rectangle, where one of its dimensions coincides in length with the distance between the guide pin and tracing point, and the other with the circumference of the graduated wheel. The circumference of the wheel is therefore divided into fifteen (15) main divisions, each division representing one square inch area of the rectangle.

Each of the main divisions are sub-divided into five (5) equal parts, each one representing one-fifth (1-5) or twenty hundredths of a square inch. The vernier scale is composed of ten (10) divisions, their combined linear distance being just equal to nine (9) subdivisions of the wheel. Therefore, as each subdivision on the wheel represents one-fifth (1-5) or twenty hundredths (20) then, accordingly the divisions of the vernier will represent eighteen hundredths (18) a difference of two hundredths (.02), consequently the vernier enables the sub-divisions to be read to one-fiftieth (1-50) or two hundredths (.02) of a square inch.

Accompanying the instrument is a nicely finished mahogany board, upon which it is mounted when in use, and an especially prepared blank card is firmly secured to the board upon which the Index Wheel travels. There is also fitted to the board a metal grooved guide I in which the guide pin slides being secured therein by the weight Q. The Clips C and K are for the purpose of securing the card to be measured in the most easy and convenient position for the operator. The angle clip C is fixed permanently to the board with its inner edge in a direct line with the centre of the groove in the guide I. The clip K is secured to a slide that is moveable in order that it may be adjusted to any length of card. The guide I is secured by a suitable thumb screw on the under side of the board; said guide is shown in the cut with its end projecting beyond the board, which is its proper position when in use. The guide may, if desired, be reversed on the board which will bring its end even with the same, and affords a better

opportunity of packing or laying away when not in use. In preparing to use the instrument the indicator card is first placed under the clips C and K which are so made as to admit of its being inserted underneath and adjusted in the proper position, that is, with the extreme left hand end of the diagram coinciding with the perpendicular edge of the clip C, while the atmospheric line is placed near to and parallel with the horizontal edge of the clip. The movable slide carrying the clip K is then adjusted, until the edge of the clip just touches the right hand end of the diagram, the presure of the clips upon the paper serving to secure it firmly while the work upon it is being performed. The slide to which the clip K is secured, is fitted so that only a slight pressure of the thumb or finger is required to move it in either direction.

The mean effective pressure of an indicator diagram being one of the principal factors in the computation of power of steam engines, hence this particular location of the diagram upon the board (as represented in the cut) is only necessary where the finding of this quantity is desired. Otherwise, in cases where only the area of the diagram or other irregular figure is needed, it may be placed in any desired position without reference to any point, and the area read directly from the graduated wheel. The instrument is then arranged upon the board with its guide pin inserted in the groove of the guide I, and secured therein by the weight Q. The tracing point O is then moved to the extreme right hand end of the diagram, where the line is in contact with the clip K (as shown at D). Here make a slight indentation in the paper, by pressing the thumb against the top of the tracing point; this gives the starting point from which to trace around the diagram

The zero mark of the graduated wheel is then turned and made to exactly coincide with the zero mark on the vernier. In commencing operations the direction in which the diagram should be followed by the tracing point is; first along the back

pressure and compression lines, thence returning by way of the expansion curve to the starting point. The only object in tracing in this direction being that the main divisions on the wheel are numbered towards the left from the zero mark, and consequently in this direction any movement of the wheel is recorded in regular order, as 1, 2, 3, etc., whereas, if the diagram is traced in the opposite direction, the reading will be the reverse of this, as 14, 13, 12, etc., and, although the circumferential movement of the wheel would be precisely the same in either case, the only consequence of tracing the diagram in the latter direction would be the inconvenience of reading the areas.

In the measurement of indicator diagrams, for mean effective pressure, no attention whatever need be given to reading the areas. After the tracing point has made a complete circuit and again reached the starting point, it is then moved upward along the edge of the clip K until the zero mark of both the wheel and vernier coincide. Another slight indentation is then made at this point, (as shown in the cut at A). The distance between these two indentations (D and A) represents the average height of the card, and also, if this is measured with a scale corresponding in pounds to the denomination of the spring with which the diagram was taken the said measurement will be the mean effective pressure of the diagram in pounds per square inch, according to the spring and scale used.

Where two diagrams are taken on the same card it is advisable to measure and find the average pressure separately; for the purpose of comparison with each other. By the use of this instrument all imperfections or irregularities whatever in the outline of the diagrams whether to be added or subtracted, are accounted for, and the final results given exact. For instance, where a loop is formed in the diagram (as in Fig. 62) caused by the expansion line crossing the atmospheric line early in the stroke, and running below to the end, thereby

dividing the diagram into two distinct parts, its outline should be traced in the same manner as is done in any well formed diagram, as the principle upon which the instrument is constructed, enables it to perform the operation of addition or subtraction with the greatest exactness and will consequently subtract the effect of the said loop from the positive part of the diagram and the reading of the instrument after the diagram has been traced, will give the net average pressure per square inch throughout the stroke.

Where the instrument is used to measure the *area* of any figure, it is only necessary to select a starting point. Adjust the zero marks on the wheel and vernier to coincide, and trace around the outline of the figure, then its correct area will be found from an observation of the number of main divisions, and subdivisions of the wheel that have passed beyond the zero mark on the vernier.

For example: Suppose upon noting the number of main divisions of the wheel that have passed the zero mark of the vernier, we find the largest figure to be three (3) which will represent inches, and the number of sub-divisions that have also passed the zero mark of the vernier, to be four (4), each subdivision representing one-fifth (1-5) or twenty-hundredths (.20) of a square inch, and the number of the division on the verneir which exactly coincides with a division on the wheel to be two (2), each representing two-hundredths (.02) of a square inch, therefore the reading taken from the instrument in this position will be $3+(4\times.20)+(2\times.02)=3.84$ square inches, as the area of figure. This instrument being of careful and delicate construction, should be handled with the greatest care and kept perfectly free from any matter that might interfere with the movement of the wheel; thereby insuring accurate results.

To a great many users of the Averager or Planimeter, shown and described in Fig. 87, it may be considered a fact that the reason and principle upon which its accuracy is based,

in the measurement of the area of any irregular figure, is, to a certain extent, shrouded in mystery; but nevertheless it is well-known that a comparison of its readings, taken from figures of known areas, prove its reliability and correctness, and can under all circumstances be depended upon for correct results; hence, a study of the theory of its operation may be interesting to many.

The manipulation of the instrument being easily performed, as before described, and its theory quite simple, we shall endeavor to make clear and explain why, by simply passing the tracing point around the outline of a given figure, its exact area will be denoted on the registering wheel. In attempting this we shall leave out, wherever possible, the use of the higher mathematics usually employed in connection with a discussion of the subject, and also shall first consider the instrument in its application to the measurement of areas, and which consideration will also apply in principle to all other planimeters. Although the principle is simple, still it is necessary for the reader not conversant with it, to follow closely the explanation in order to become familiar with the peculiar movements and actions of the registering wheel.

In Fig. 88 the outline A, A1, C, D and A3, is assumed for our purpose to represent an indicator diagram, the expansion curve being shown in dotted lines. The point A on the diagram is selected as the starting point from which to trace and consider the figure. Probably a better idea may be had, by first confining our study of the subject to a part of the diagram; that is, the area of the square inclosed by the lines A, A1, A2 and A3. The line A1-B1 represents the arm of the instrument, and shows its position after the tracing point has been moved horizontally away from the starting point A to the position at A1. This line (A1-B1) may be assumed to be a small round rod, the end (B1) being guided to always move in a straight line, as shown by dotted line from B1 to B3; while

the end (A1) carrying the tracing point is free to be moved in any direction, and to any point of the diagram. This arm or rod is also shown here as the axis of the registering wheel W, and is represented in this way for the purpose of simplifying matters in the way of its demonstration, and because the final results of the registering wheel W, in having its axis coincident with the arm A1-B1, will be precisely the same as though its axis was located at a distance from, and parallel to, a line drawn between the tracing point A1 and guide pin at B1; (this latter construction being that of the instrument represented in Fig. 87). In the diagram the four positions of the registering wheel are shown at Wo, W, W1 and W2, when the tracing point is respectively at each corner of the square or rectangle A, A1, A2 and A3.

Suppose the tracer to be at the starting point A—the arm then coinciding with line A-B, and the registering wheel at Wo; now, any movement at the tracing point up or down on that line will simply cause the wheel to slide in the same direction, but without causing any rotation of it, because its axis is coincident or parallel to said line; hence, any movement of the wheel in a direction parallel to its axis will not cause rotation.

If, now, the tracing point be moved to the right, in a horizontal direction, until the arm is in the position A1-B1, then the wheel has moved from Wo to W, and has revolved a certain distance at its periphery; the amount depending upon the degree of the angle formed by its departure from the starting point, or line A-B; but, any consideration of the amount of this movement, as a factor in the case, is wholly unnecessary, because all motion imparted to the wheel by its departure from the line A-B is always exactly cancelled (in whatever direction it may return) on arriving at the starting point.

Now, consider the action of the wheel in an assumed downward movement of the arm, from its position A1-B1 to a

position A2-B2, then the area swept over by the arm is the space A1-B1, A2-B2, which is exactly equal in area to that of the square, or rectangle A, A1, A2 and A3, because they both have the same base, and between the same parallels. The result of this movement of the arm will cause the registering wheel to move from W along the dotted line to the position W1. The wheel being in contact with the paper, and the direction of its movement from W to W1, at an angle with its axis, will consequently cause it to revolve while being moved downward to W1, and the amount of this motion at its periphery will be represented by the line W-N, which is the sine of the angle W, W1, N, with W1-W as the radius of the circle; consequently, by knowing this angle, the length of its sine can be found from a table of sines, in which their lengths are given for any angle, the radius being unity. Suppose, for example, the angle formed between the arm A2-B2, and the vertical line of movement of the wheel from W to W1, to be one of $18\frac{2}{3}$ degrees, and that the vertical distance traversed by the wheel to be two (2) inches; then from a table we find the sine for that angle to be .32 in terms of inches, at unity. Therefore, if this be multiplied by the vertical movement, two (2) inches ($.32 \times 2 = .64$ in.), this product will be the length of the line W-N.

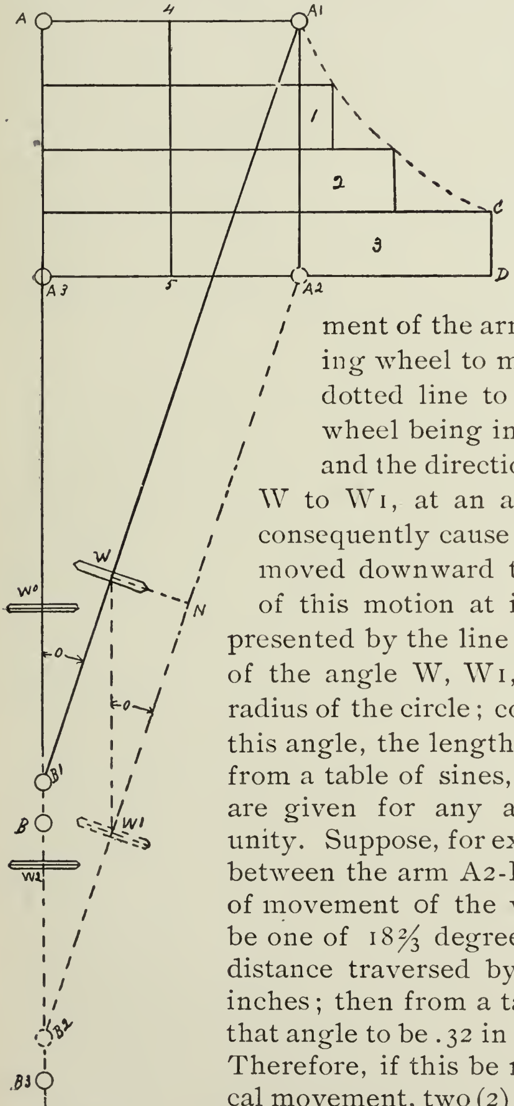


FIG. 88.

The rotary motion, therefore, that has been imparted to the periphery of the wheel (by being in contact with the paper over which it runs), in moving from W to W1, is equal to the length of the line W-N, and the final result in the readings are precisely the same as though the wheel was first rolled in the direction of its rotation upon the line W-N, and then without rolling moved down the line A2-B2, coincident with its axis to W1; hence, this distance (W-N) if multiplied by the length of the arm A1-B1 (W-N$\times$A1-B1), is equal to the area of the space passed over by the arm, and is also equal to the square or rectangle A, A1, A2 and A3, for the reason before stated, both having the same base, and between the same parallels.

Suppose now the tracing point to be moved from its position A2 to A3; in doing this the wheel will have moved from its position W1 to W2, and revolved at its periphery an amount exactly equal to that which it revolved in being moved from A to A1. Therefore, as the wheel has revolved through the same angle in both cases, but the motion being in opposite directions, will thereby cancel each other, and, as a consequence, leave no positive results on the registering wheel.

In moving upward from A3 to the starting point at A, the wheel will not revolve, because this movement is parallel to its axis.

From a careful consideration and study of the matter, the following facts may be deducted and readily observed:—1st. That all movements of the tracing point that are in a direction parallel to the axis of the wheel, will not cause it any revolution. 2d. That all revolution of the wheel caused by a departure of the tracer from the starting point A, will be exactly cancelled on its return to that point. 3. That the only permanent record remaining on the wheel (after tracing the diagram), is derived entirely from vertical movements of the tracer relative to the starting point A, the amount of its revolution depending upon the angle formed by the arm with the line

A-B, together with the vertical distance passed over by the tracer. 4th. That the amount of revolution of the wheel is always equal to the sine of the angle (formed by its axis with the direction of its movement), multiplied by the distance of its vertical movement. It also becomes evident that any departure of the tracer A1 from the starting point A, is proportional to the sine of the angle so formed, multiplied by the length of the arm A1-B1. Supposing this angle to be one of $18\frac{2}{3}$ degrees (the same as the previous example), the sine of which at unity is .32 inches, and the length of the arm A1-B1 to be 6.25 inches, then ($.32 \times 6.25 = 2$ inches), the distance of the tracer's departure from A. This product, then, if multiplied by any vertical movement of the tracer, is equal to the area corresponding to such movement.

If in place of moving the tracer to A1 it had been stopped at the point 4, and moved down the line to 5, and thence through A3 to the starting point A, the area of this rectangle recorded on the wheel would be just one-half that of A, A1, A2 and A3, because this angle is proportionately one-half of the former, thereby making it more acute, and consequently its sine would also be only one-half. The effective rotary motion of the wheel in this case is derived entirely from its vertical movement along the line from 4 to 5. Therefore by applying the rule (A1.B1 $\times$ sine of angle $\times$ vertical movement = area), it will be found that the area of the latter is only one-half that of the larger rectangle.

We may now consider the whole of the diagram, Fig. 88, including the dotted curve line A1, C.; and in doing so it must be understood that whatever has been said in reference to the larger rectangles, is equally true of any, and all, others that may be inscribed within its extreme outline; (however numerous, whatever their size, and wherever located). If in addition to the larger rectangles we inscribe smaller ones, numbered 1.2, and 3 in the diagram, and from the starting point

A, carry the tracer around their extreme outline, so as to include their measurements; it will be found on returning to the starting point, that the reading on the wheel will be increased by an amount just equal to the combined areas of the smaller rectangles 1-2 and 3, (over the readings taken from the larger ones contained in the square A-A1-A2 and A3:) and such reading will be *approximately* the area of the diagram.

Other additional smaller rectangles may still be inscribed in the spaces adjacent to the dotted curved line; (said space yet remaining unaccounted for in the reading;) And if these smaller rectangles should be made sufficiently numerous, and measured separately, by passing the tracer, in turn, around the extreme outline of each, without removing from the paper, the wheel would mechanically add the area of these rectangles together, and on the return of the tracer to the starting point A, the reading of their combined sum would be *exactly* equal to the area of the whole diagram; and which will also be equivalent to the reading that would appear on the wheel after passing the tracer around the outline of the diagram, including the curved line A1, C.

Therefore it will be seen that as the horizontal movements of the tracer, over the lines of any rectangle, cancel each other, while only vertical movements relative to the starting point A, leave any permanent record on the wheel, the results will be precisely the same as though the tracer had simply moved over the actual extreme outline of the diagram

As the area of any rectangle or parallelogram, is equal to the product of its length multiplied by its vertical height, it is readily seen, that after passing the tracer once around the outline of any figure, (regular or irregular;) its exact area will be denoted by the number of divisions of the wheel that have passed the zero mark on the vernier; consequently the necessary height of a rectangle of the same length, to contain the same area may be determined by moving the tracer upward along the clip K. until the same number of divisions have returned; and the zero marks on the wheel, and vernier again coincide,

(as mentioned in the description of the instrument in Fig. 87), which point will be the average height of a parallelogram, containing exactly the same area as the figure traced. Therefore in case the figure traced be an indicator diagram, such point *must be* its average height, because it is the height of a rectangle of the same length, and same area as the diagram. Consequently if this average height be measured by the scale of the spring with which the diagram was taken, the units of the scale will show the *mean effective pressure* in pounds per square inch throughout the stroke of the engine without any computation whatever.

The number of parts into which the circumference of the wheel may be divided; can be 10-15-20 or any other convenient number; the only requisite being, that the circumference of the registering wheel shall be so proportioned that one complete revolution on its axis, would cause it to roll over the paper an amount, such, that if this distance was multiplied by the length of the tracer arm, that their product will be equal to the area of a rectangle containing *exactly* some *whole number* of square inches; (suppose for example fifteen:) now if the circumference of the wheel be graduated into the same number of divisions, and numbered accordingly from one to fifteen, then each division on the wheel will represent one square inch: and the area of any figure, after the tracing point has passed around its outline, will be indicated in square inches, according to the number of divisions of the wheel that have passed a given point. We again mention the necessity of handling the instrument with great care in order to obtain the most accurate results, never allowing the contact edge of the registering wheel to become indented in the least, keep perfectly free from rust or deposit of any kind, and a good plan, is to pass a thin piece of paper between the wheel and vernier to remove any particles of dust that may accumulate and destroy the free action of the wheel; see that the spindle runs easily, and freely between the centres, and practically without end play; and care must also be taken to prevent the tracer arm from getting bent or in any way out of shape.

The Amsler Polar Planimeter. Illustrated in Fig. 89 is another device for measuring diagrams: This instrument does not give the mean effective pressure directly, but it determines the area of the diagram, and from this area the mean effective pressure is computed in the manner described. It consists of the two arms, A and D, and the measuring wheel, C. To operate it, a piece of smooth, hard paper is laid on the table, and the instrument placed upon it, with the needle point, A, pressed into the board. This point serves as a centre about which the apparatus is turned. The indicator card is laid under the tracer, B, and held either by tacks, which fasten it to the table, or what is quite sufficient, by the pressure of the fingers. The tracing point is set on the line of the diagram—say near the middle of the steam line—and a slight indentation made in the paper, to serve as a starting

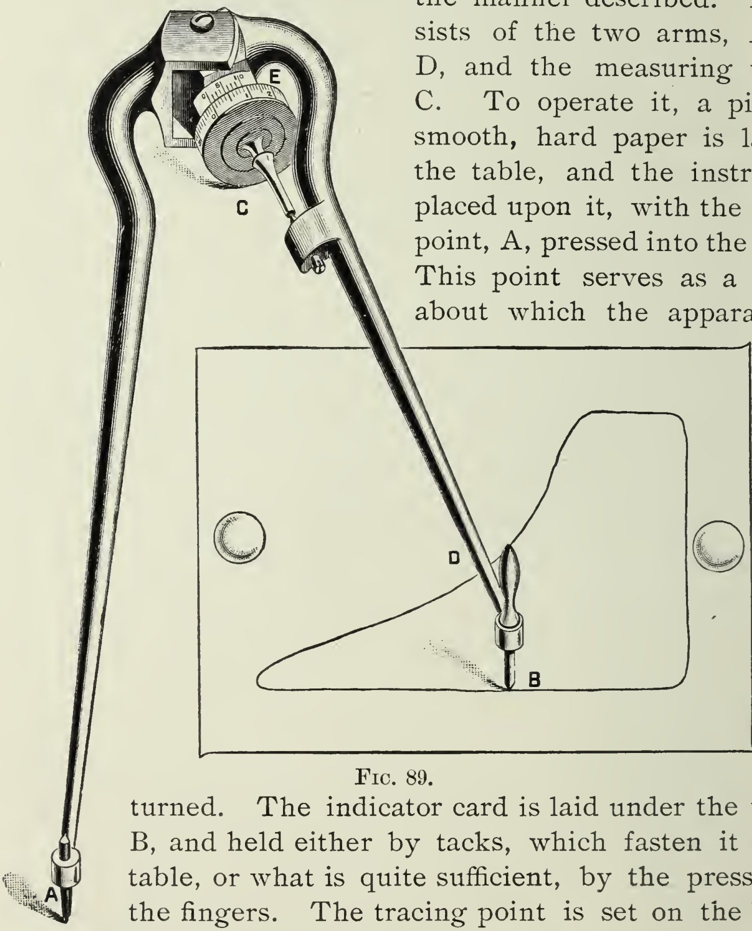


Fig. 89.

point. The graduated wheel is set at the zero mark. The tracer is then moved over the line of the diagram in the direction of motion of the hands of a watch, finally making a complete circuit, and returning to the starting mark. The number of divisions and fractions of a division shown on the wheel at the point opposite the stationary zero mark, indicates the area of the diagram traced. The wheel has ten main graduations, each of which represents one square inch of area. Each main division is subdivided ten times, and each subdivision represents one-tenth of a square inch of area. A stationary vernier scale E is placed beside the graduated edge of the wheel, and serves to indicate the smaller fractions, viz., hundredths. To read the vernier, the eye is run along the stationary scale till a line of division is found which is just opposite a division on the wheel. The number of the division on the vernier, reckoned from the zero mark, is the number of hundredths sought. If, for example, the reading of the area is two main divisions on the wheel, and four of the subdivisions and the line of coincidence on the vernier is number seven reckoned from zero, the area sought is 2.47 sq. in.

To reduce this to mean effective pressure, two perpendicular lines are drawn, one through each terminal point of the diagram, and the length of the diagram is measured by measuring the distance between these two perpendiculars. Suppose this distance is 3.78 in., and suppose the number of the spring employed is No. 40. Then the mean effective pressure is found by dividing 40 by 3.78 and multiplying the result by 2.47. Making this computation the mean effective pressure sought is 26.13 lbs. per sq. in.

In working the Polar Planimeter, care must be observed to place the diagram so that the two arms are not brought too near each other at one end of the course, nor yet carried too far apart at the other end. At a point midway between the two extremes the two arms should lie about perpendicular to each other.

As an example of the manner of computing the horse-power of an engine, suppose an engine has a cylinder 15 inches in diameter, a piston rod 2½ inches in diameter, a stroke of 2½ feet, running at a speed of 135 revolutions per minute. Suppose the indicator diagrams show a mean effective pressure of 36 lbs. per square inch, this being the average of the indications at the two ends. The area of the cylinder to be used is the net area obtained by deducting one-half the area of the rod. The area of a cylinder 15 inch in diameter is 176.71 square inches.

One-half the area of a rod 2½ inches in diameter 2.45 square inches. The net area to be used in the computation is $176.71-2.45=174.26$ square inches. The speed in feet per minute is $135\times 2\frac{1}{2}\times 2=675$ feet. The horse-power developed, therefore, is

$$\frac{36\times 174.26\times 675}{33{,}000}=\frac{4{,}234{,}518}{33{,}000}=128.3 \text{ H. P.}$$

That is

$$\frac{\text{M.E.P.}\times\text{net area of cyl. in sq. ins.}\times\text{piston speed in ft. per min.}}{33{,}000}$$

When the engine has more than one cylinder, the power developed in each cylinder is computed in the manner given, and the results added together.

CHAPTER XXIV.

COMPARISON OF DIAGRAMS FROM THROTTLING AND CUT-OFF ENGINES.

The following diagrams in this chapter were taken from what originally was, (when new) a pair of side by side non-

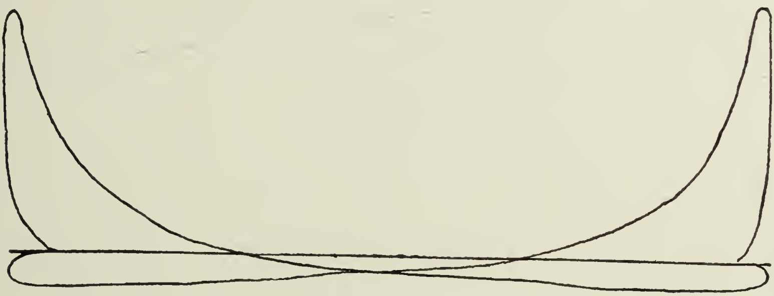
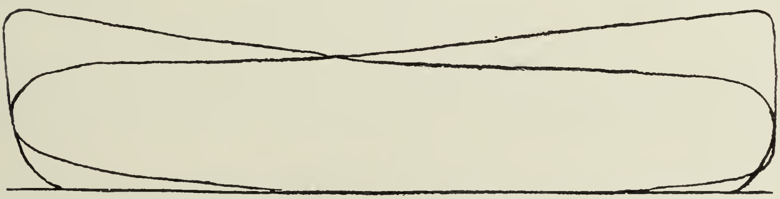


Fig. 90.



Extreme range of New Valve gear.

Fig. 91.

condensing throttling engines, and connected to the crank shaft at right angles, or quartering.

After running in this manner for some time it was deemed advisable, and also as an experiment, to convert the valve motion of one of the pair into a special automatic cut-off valve gear in order to use the steam expansively on one engine, and throttling on the other.

This alteration of the engine being completed, the diagrams both in Fig. 90 and Fig. 91 in connection, were taken to show the extreme range of this new valve gear.

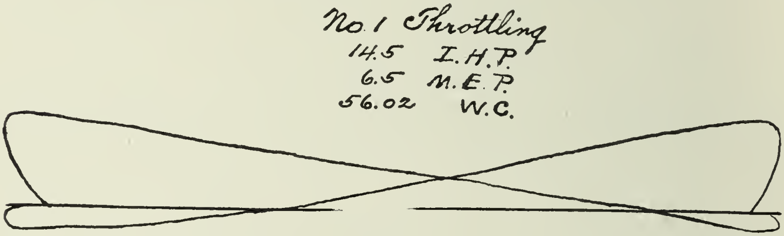


Fig. 92.

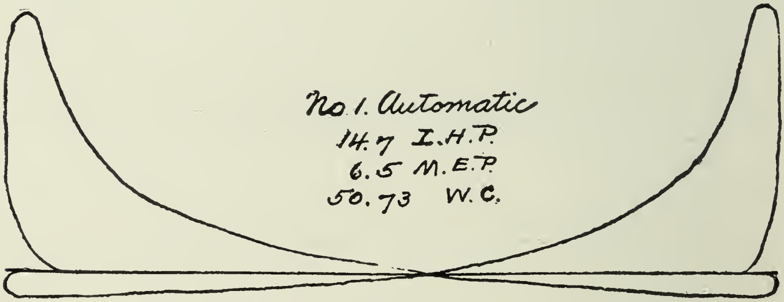


Fig. 93.

The diagrams from Fig. 92 to Fig. 103 inclusive, are facsimilas of diagrams taken in pairs, after this alteration had been made on the engine; having one cylinder throttling, and the other with automatic valve gear, and were taken to illustrate the difference in the two systems of government at different loads; the data of the engine being as follows:

AUTOMATIC ENGINE		THROTTLING ENGINE	
Dia. of Cylinder,	$15\frac{3}{16}$ in.	Dia. of Cylinder,	$15\frac{3}{8}$ in.
" " Piston Rod,	$2\frac{3}{8}$ "	" " Piston Rod,	$2\frac{3}{8}$ "
Length of Stroke,	42 "	Length of Stroke,	42 "
Revolutions per min.	60	Revolutions per min.	60
Piston speed " "	420′	Piston speed " "	420′
Scale of Spring,	40	Scale of Spring,	40

Thus giving a value of the engine constant on the automatic side of 2.275, and of the throttling side 2.334, indicated horse power, for each pound of mean effective pressure, at a speed of 60 revolutions per minute.

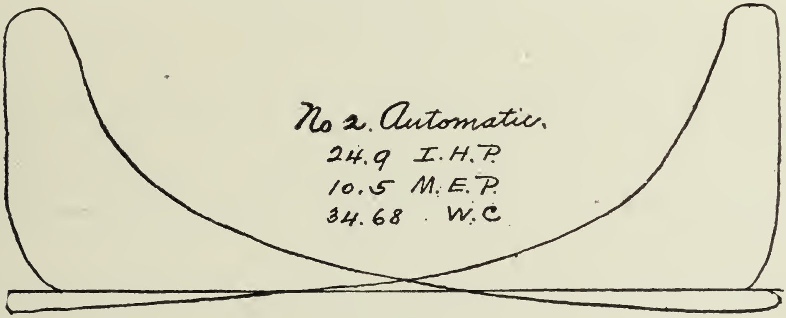


FIG. 94.

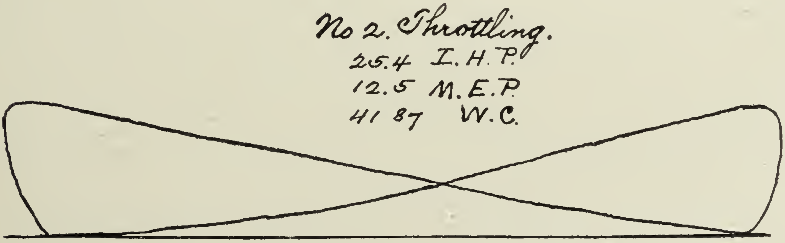


FIG. 95.

In calculating the horse power from the diagrams, the factor that has been used to represent speed has been the *actual number of revolutions* of the engine, at the time each diagram was taken.

The mean effective pressure was obtained by the use of a planimeter, and the steam or water consumption, computed by the formula $\frac{859375}{\text{A. T. P.} \times \text{M. E. P.}}$ that is, by dividing the constant 859375, by the volume of the steam at absolute terminal pressure, multiplied by the mean effective pressure in pounds per square inch of the diagram; the quotient being the number of pounds of steam or water consumed per hour per each indicated horse power developed by the engine.

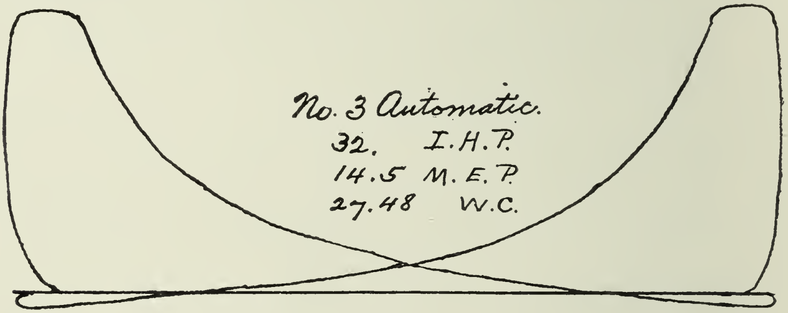


Fig. 96.

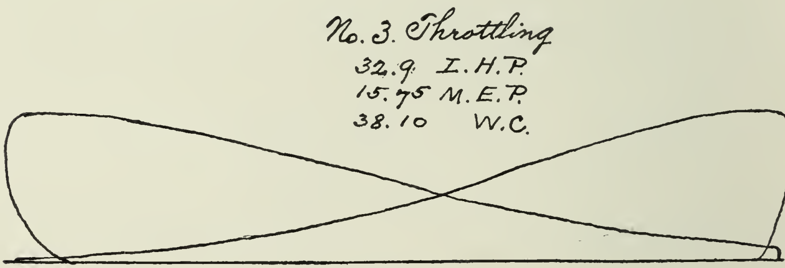


Fig. 97.

In the latter calculation however, no allowance has been made for compression and clearance; the rule for making such allowance being described in connection with Fig. 77, Chapter XXI

These diagrams were selected from a large number, with two objects in view; first to show the different forms or outlines of the pressure areas between the automatic, and throttling

system of governing, from pairs of diagrams of approximately the same horse power; and second, to show the relative rates of water or steam consumption of each, in pounds per horse power per hour.

In addition the diagrams from the automatic side also show that the minimum amount of steam consumption (and

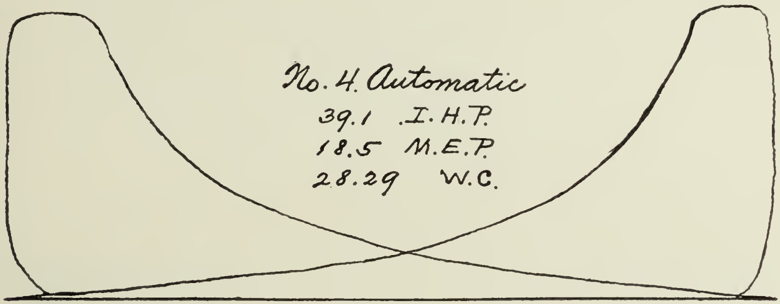


Fig. 98.

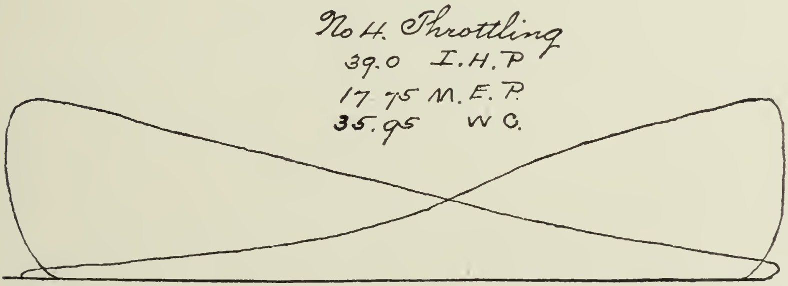


Fig. 99.

consequently the greatest economy) occurs in diagrams No. 5, Figs. 100 and 101, where the cut-off takes place at about twenty-two hundredths of the stroke, corresponding to about 4.57 ratio of expansion, and to very nearly the generally considered most economical point of cut-off in automatic engines working at a steam pressure of from 60 to 75 pounds pressure per square inch.

By reference to the diagrams from either the automatic or throttling engine it will be observed that a gradual decrease in the water consumption takes place as the power developed by the engine increases, or, as in case of the automatic engine, when the cut-off takes place later in the stroke.

The minimum consumption appears to have been reached

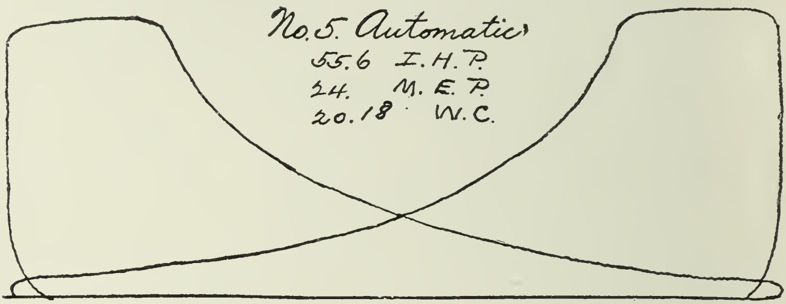


Fig. 100.

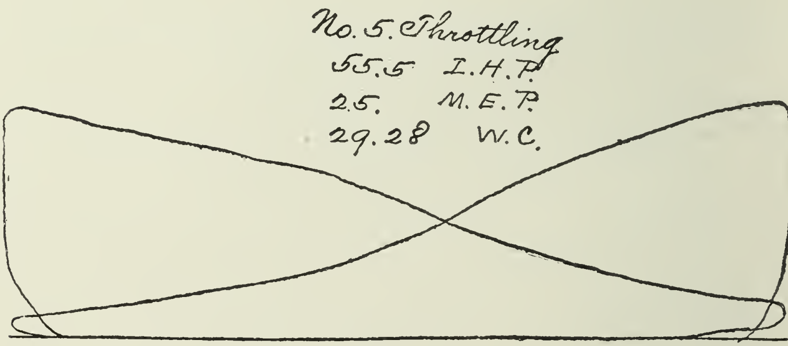


Fig. 101.

in either case, in the diagrams Nos. 5, and it will be seen that in all diagrams on either side whether developing more or less power, that an increase in the rate of water consumption invaribly takes place.

In these diagrams the economy of the automatic cut-off engine, over that of the ordinary throttling system is very

fairly exhibited, and shows the relative economy that might be expected from the automatic, under ordinary circumstances, over that of the throttling system of regulation.

The outlines of the various diagrams of each system are fair representations of what ought to be expected from an engine with properly designed valve gear, and with evenly and accurate adjustments of the same.

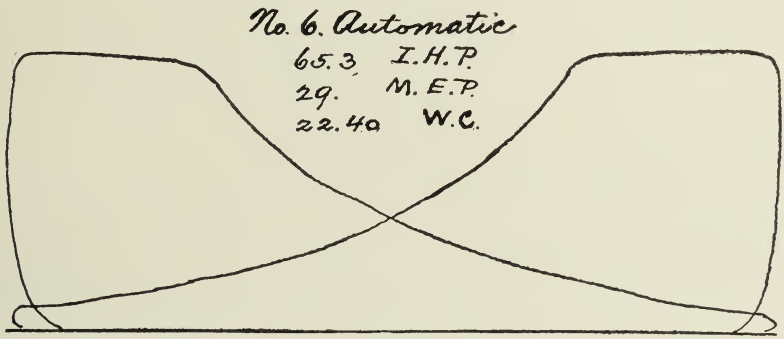


FIG. 102.

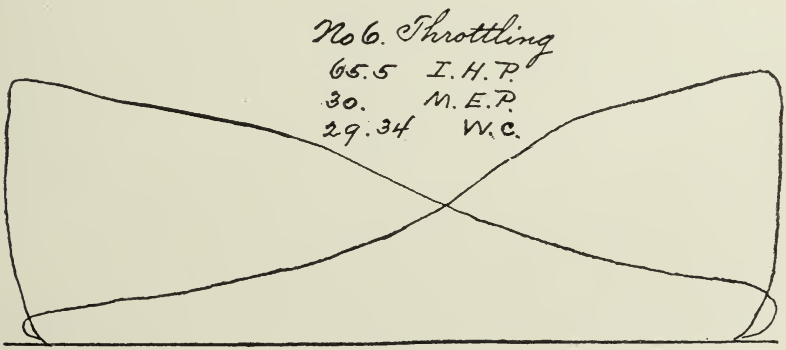


FIG. 103.

It must be understood that the diagrams here presented are not intended, or claimed, as representing the highest efficiency and economy obtainable, but are only given from actual ordinary practice, in order to be able to make an interesting comparison between the two systems of regulation.

CHAPTER XXV.

ECONOMY OF EXPANSION.

As regards the economy of expansion it will be found by reference to the Table No. 8 on the properties of saturated steam that the weight of a given volume of steam varies very nearly in proportion to its pressure that is, a cubic foot of steam at 60 pounds pressure weighs approximately twice as much as the same volume at 30 pounds pressure.

A given weight of steam represents the same weight of water that must be evaporated to produce it, consequently the lower the terminal pressure at the end of the stroke in a given cylinder the less will be the amount of water exhausted as steam; and as the measure of the work done in the cylinder by the steam, is proportional to the mean effective pressure, it becomes evident that economy in the use of steam in an engine, consists in getting a high mean effective pressure, in connection with a low terminal pressure.

This can only be accomplished by cutting off the supply of steam, at a time when the piston has only completed a part of its stroke, and by that means obtain an additional amount of work from the expanding steam to the end of the stroke.

Suppose in a steam cylinder using steam at 100 pounds absolute, the steam be cut off when the piston has moved one-fourth of its stroke, then the mean pressure calculated according to the rule in Chapter 18 will be almost 60 pounds; and

this is obtained by using only one-quarter of a cylinder full of steam; whereas if the entire cylinder had been filled the mean pressure would have been only 100 pounds, with an expenditure of four times the quantity of steam used as when cutting off at one-quarter stroke.

This indicates the direction in which a saving is effected; but in practice however, condensation in the cylinder, and other causes prevent this full theoretical gain from expansion being realized; at the same time however the gain from this source is an important one.

Therefore from what has been said in this connection it would be natural to expect, in considering diagrams from first-class cut-off engines, to find the initial pressure of the diagram high, as compared with the boiler pressure, also with straight or fairly straight steam lines, and sharp cut-off; because these all tend to bring about both high mean effective, and low terminal pressure; also whatever tends to make the terminal pressure higher than it should be, represents waste of steam.

Economy of High Pressure. It is a well established fact that the use of steam of a high pressure tends to greater economy in the engine; the reason for which is simply as follows:

Suppose in this case an engine working without expansion, and in order to simplify the matter, assume that there is no clearance. Assume also the engine to be working non-condensing, with an absolute back pressure of 15 pounds, or three-tenths above the atmosphere.

If steam of 20 pounds absolute pressure is used in the cylinder, the mean effective pressure on the piston is $20-15=5$ pounds.

If the piston has a total displacement of one cubic foot, then we are using one cubic foot of steam, of a pressure of 20 pounds at each single stroke of the piston.

In the Table No. 8 on the properties of saturated steam, the weight of a cubic foot of steam of this pressure is found to

be .0511 pound, and the heat units per pound 1183.5; hence the cubic foot of steam from which the mean effective pressure of 5 pounds has been obtained, contained $1183.5 \times .0511 = 60.47$ heat units.

Now instead of steam of 20 pounds pressure, suppose steam of 100 pounds absolute pressure be used: Then the mean effective pressure would be $100 - 15 = 85$ pounds.

The weight of a cubic foot of steam of 100 pounds pressure is .2330 pounds and one pound contains 1213.8 heat units. Then as before $1213.8 \times .2330 = 282.8$ heat units used.

In the first case 60.47 heat units have been used, equal to $60.47 \div 5 = 12.09$ heat units for each pound mean effective pressure; whereas in the second case $282.8 \div 85 = 3.32$ heat units for each pound mean effective pressure have been used.

This great difference and discrepancy in the two instances, arises from the fact that the greater part of the total heat of the steam still remains in it at the pressure of the exhaust, (15 pounds) and all of this heat is entirely lost with the exhaust steam.

The only heat that can be converted into useful work, is the quantity that is added above exhaust pressure; therefore it becomes apparent that a greater per cent of the total heat can be made available, as the pressure increases.

From this it must be clearly understood, that a large proportion of the heat that enters into the steam used in a steam engine is requisite in order to raise it to the pressure at exhaust; (15 pounds), a pressure at which *no work* can be utilized from it.

The diagram Fig. 104 fairly illustrates the economy of the use of high pressure steam in a cut-off engine.

The full lines of the figure represents an actual diagram, and above it the shaded portion has been drawn by hand to show precisely as if the whole of the diagram had been taken at a higher steam pressure.

The pressure of the steam at the end of the stroke, or after having performed *all of the work* there is in it, is not changed; as the terminal pressure T remains the same: hence we conclude that as far as we can judge by the diagram, the work represented by the shaded portion could be done without additional expense.

This is on the assumption that steam expands according to the Mariotte law, but if the result were calculated from Table

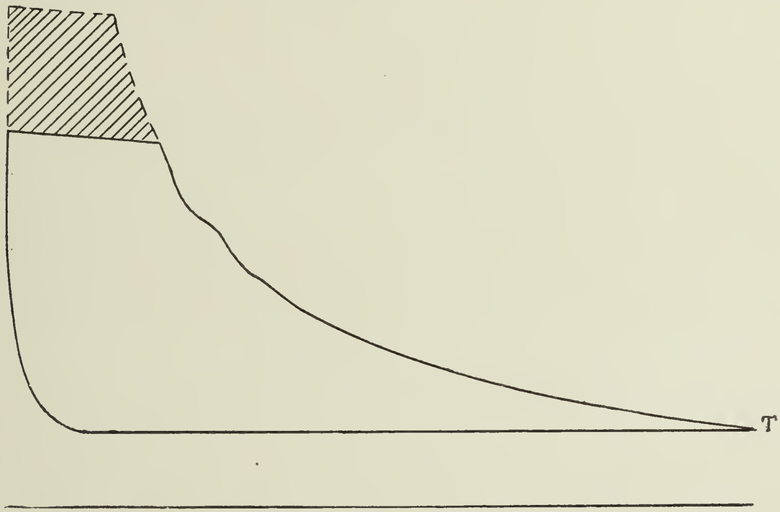


Fig. 104.

No. 8 (properties of saturated steam) it would be somewhat different, but we have no positive assurance that it would be any more correct; therefore for all practical purposes, this presentation of the subject should prove acceptable.

There is a point however where it is the reverse of economical to run at very high pressures, and that is when the load on the engine is so light, and cut-off so short, that the steam expands below the atmosphere, and thereby forming a long loop in the diagram as illustrated in Fig. 54 and 62.

In such cases the engine is running a portion of each stroke on the momentum of the moving parts.

Where this occurs it is better to reduce the boiler pressure until expansion shall come quite down to, *but not below* the atmosphere.

In such an instance a considerable advantage is realized by reducing the speed of the engine; as the resistance of the atmosphere must be overcome in running an underloaded, as well as in a heavily loaded engine; therefore by reducing the speed this factor is made less, and on lightly loaded engines becomes a considerable percentage of the total power; this percentage decreasing in extent as the engine is more heavily loaded.

In connection with losses from having under loaded engines, is the fact also, that the friction of the engine, condensation, etc., are all present almost regardless of the load, and always appear in a greater proportion in light, than in heavy loads; therefore the final expense is decreased when running at a slower speed; as there is a better opportunity to secure a later cut-off, since the power being the same, with reduced speed, the mean effective pressure must be proportionally increased.

CHAPTER XXVI.

THE POINT OF CUT-OFF.

It must *not* be understood, (in reference to steam expansion) that the higher the grade or ratio of expansion, the greater is the economy, because, quite the reverse may often happen, as the results in practice are greatly modified in various ways by other considerations.

Consequently the most economical point of cut-off will vary in different types of engines, in accordance with the special conditions, and circumstances to which each may be subjected; such as high or low initial pressure, condensation of steam in the cylinder, different methods of jacketing and superheating, amount of back pressure and compression, incorrect adjustment of the valves, also a size of engine that is not adapted to the work, etc.; any or all of which necessitates a change more or less of the point of cut-off, in accordance with any particular condition, in order that the greatest efficiency of the engine may be realized.

Initial Pressure. In engines with an early cut-off, and where a *high rate* of expansion prevails, the mean or average pressure throughout the stroke must necessarily be *low;* and a low mean pressure, for a given power necessitates the use of a larger engine.

Also a high rate of expansion leads to a low terminal pressure, and to expel the steam from the cylinder after it has

performed its work, will require from one to three pounds pressure above the atmosphere; therefore if the rate of expansion be such that the terminal pressure falls below this, the expansion is excessive, and the reverse of advantageous.

In non-condensing engines the lowest possible terminal pressure coincides with the pressure of the atmosphere, or about $14\frac{7}{10}$ pounds per square inch absolute; but 18 pounds may be considered as the lowest pressure to which steam can be expanded to *advantage;* and where the exhaust passages are small and tortuous, or where the exhaust steam contains considerable moisture, a still higher terminal pressure will be more economical.

In condensing engines, the temperature in the condenser is usually about 100 degree Fah., and which corresponds to a pressure of very nearly *one pound* to the square inch; but the presence of air in the condenser generally prevents the pressure falling *below two pounds* per square inch.

From three to four pounds is the more usual and may be considered as the lowest advantageous final pressure, to secure the best economy. The highest advantageous rates of expansion with jacketed cylinders appear in practice to be between *twelve* and sixteen; but in unjacketed or exposed cylinders, the limit of advantageous expansion is much below the lowest of the rates mentioned.

The principal cause of the discrepancy between the theoretical, and actual economy, is in the amount of *heat lost* in changing the water in the cylinder from a heated liquid state, to the condition of steam; most of this *heat* passing out with the exhaust steam, either into the condenser or the atmosphere. It has been fairly well demonstrated by frequent tests, that the most favorable point of cut-off, in a simple non-condensing engine, is usually about one-fourth stroke, when using steam of from 80 to 90 pounds per square inch initial pressure.

Where the cut-off is earlier than this a greater per cent of loss is induced through cylinder condensation; and where the cut-off takes place later in the stroke there is also an increase of loss, from the fact that the steam is exhausted and lost while yet at a considerable pressure.

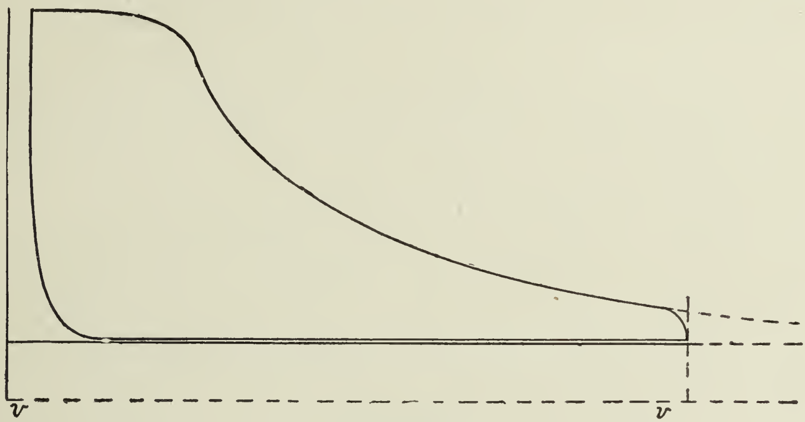


Fig. 105.

Fig. 105 is a fair representation of an indicator diagram from an engine working under the favorable conditions mentioned.

The results of some recent experiments on a 17 by 30 inch stroke non-condensing double valve engine by Prof. J. E. Denton, to determine the relation of steam consumption to point of cut-off is illustrated by the graphical diagram Fig. 106.

The line A. B. represents the stroke of the engine, which is equally divided into tenths and twentieths, to denote the various points of cut-off in fractions of the stroke, or periods of admission of steam to the cylinder.

From each of these fractional parts of the stroke perpendicular lines are erected, and also above these points are horizontal lines laid off 1-5 of an inch apart; each representing 4

pounds, or a scale of 20 pounds per inch; being the actual pounds of water used per horse power per hour.

Through these lines three curves are plotted, being the result of experiments at various points of cut-off, and boiler pressure by which the change in the rate of water consumption, corresponding to changes in point of cut-off, (as well as changes in boiler pressure) are readily observed.

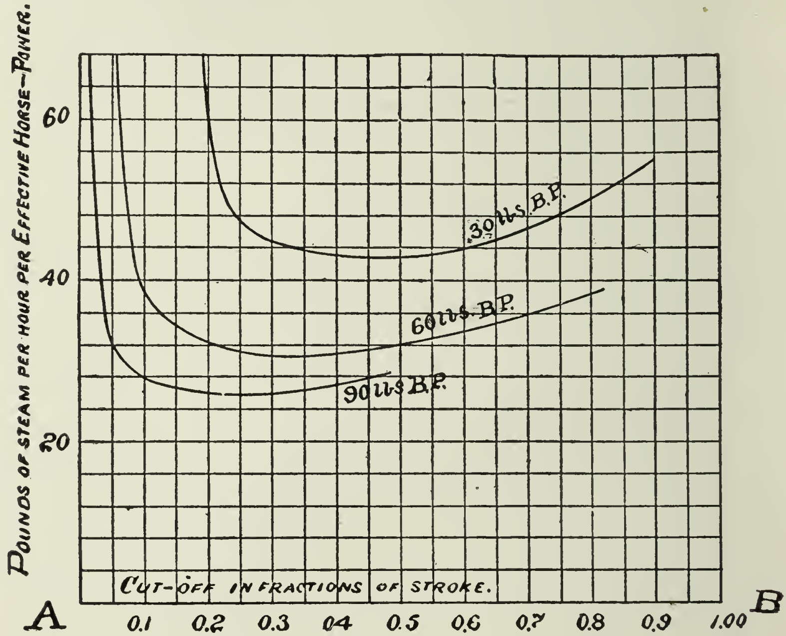


Fig. 106.

The three curves shown in the diagram corresponds to 90, 60, and 30 pounds boiler pressure respectively per square inch, and the curves were constructed by locating points as a result of tests in each case at different rates of cut-off. It will be clearly seen that the curve corresponding to 90 pounds per square inch boiler pressure and the steam cut-off at $\frac{1}{4}$, or $\frac{25}{100}$ of the stroke gave the best results, and represents a water consumption of about 26 pounds per horse power per hour; that for

60 pounds pressure, cut-off at $\frac{35}{100}$ of the stroke a consumption of about 32½ pounds, and that for 30 pounds pressure, cut-off at $\frac{45}{100}$ of the stroke; corresponds to a consumption of 43 pounds of water per indicated horse power per hour.

For a short distance on either side of these points of cut-off, the amount of water consumption is only slightly increased and the economy is practically the same, but in case of very early or very late cut-off the amount of water consumed is considerably increased; the relation of economy to cut-off being well represented by these experiments in the diagram.

In some similar experiments with a small (7x14) Buckeye automatic non-condensing engine, with a boiler pressure of 75 pounds, the same experimenter has found the best rate of consumption to be 30 pounds of water at about ¼ cut-off; and while the most favorable point of cut-off is the same in small as in large engines, the matter of economical use of steam is in favor of the larger engine and high pressure.

It should be understood that these results were deduced from refined tests, extending over a considerable period, the engines being in perfect condition as to leakage, etc., and the steam collected, condensed and carefully weighed after the work was performed, thus eliminating errors as far as possible.

The steam was also practically free from moisture, as, in the tests of Fig. 106 it contained less than one per cent.

From these and other similar tests we are enabled to formulate the following approximate rule for best cut-off.

Divide 100 by 42 times the square root of the initial pressure and the quotient will represent the most favorable point of cut-off, expressed in fractions of the stroke. The initial pressure must be reckoned from a vacuum for condensing and from the atmosphere for non-condensing engines.

In a test for ascertaining the economy of engines, it is very important that the quality of the steam used, or in other words the amount of moisture it contains, be determined.

This quantity may be approximately found by making what is known as a calorimeter test, as explained in Chapter XXX.

There are a number of instruments of varied construction and principle, on the market for this purpose, with full and complete instructions for their use.

In the absence of such a test, no accurate conception of the amount of steam actually used by the engine to produce a horse power per hour, can be obtained, for the steam may contain a large amount of ineffective moisture in the form of spray, which can only be detected either by an application of the calorimeter, or by collecting and weighing the exhaust.

The generally acknowledged economy, that has been attained in the steam engine within the last thirty years, (estimate between thirty and forty per cent.) may be considered due, principally to the employment of multiple cylinder engines in connection with the best modern method of jacketing: also an increase of steam pressure, and superheating the steam; either or all of which (when judiciously made use of) tends to better efficiency, and more economical results.

CHAPTER XXVII

BACK PRESSURE AND COMPRESSION.

Back Pressure may be considered as one of the most serious losses in an engine, and represents the power expended in expelling the exhaust steam, and in compressing steam into the clearance spaces, and is usually called the loss from back pressure, for it must be clearly understood that the direct steam does work in expelling the exhaust from the engine, as well as in driving the machinery to which the engine is connected.

The whole of the back pressure cannot be removed, but a large proportion of the present losses would be avoided, if the *clearance spaces were reduced* to the smallest practical dimensions.

The following diagrams Figs. 107 to 110 will serve to illustrate the economy to be derived where small clearances prevail.

The diagrams referred to, are theoretical, but in each case the assumed clearances (Fig. 107 being seven per cent. and Fig. 109 being only two per cent.) are exactly with many of those found in different makes of slow speed engines.

To obtain the maximum economical results with an engine at any given cut-off, compression should be carried up to, or very nearly the initial pressure of the cylinder.

Where the clearances are large however, this is not always possible, especially with condensing engines, and without much consideration of the subject many builders of engines adjust the valve so that compression begins at about seven-eighths of the stroke.

In the diagrams the various compression lines are shown, and the results compared. The initial pressure is 100 pounds above zero in all, and the cut-off takes place at one-quarter stroke in each, so that when compression is carried up to initial pressure, the same quantity of steam, practically, is used per stroke in each engine irrespective of clearance, but the net power developed under those circumstances will differ very materially.

The exhaust lines in the non-condensing diagrams are 0.3 pounds above atmosphere or 15 pounds above zero, and in the condensing diagrams 3.5 pounds above zero, or 11.2 pounds below atmosphere.

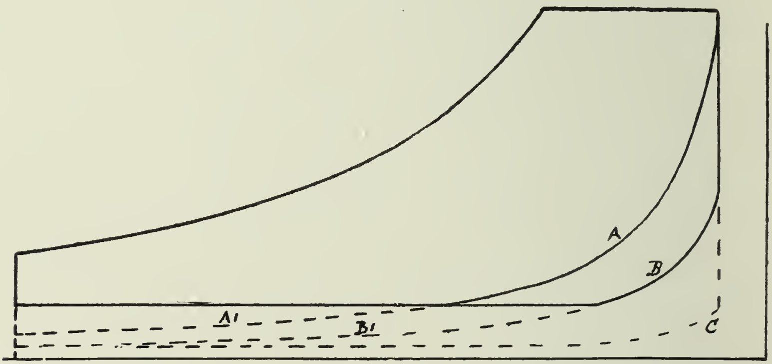


Fig. 107.

THEORETICAL DIAGRAM WITH 7 PER CENT CLEARANCE.

In Fig. 107 the solid lines represent diagrams from a non-condensing engine with compression A to initial pressure, and also compression B to 50 pounds, or one-half the initial pressure.

The dotted lines represent the back pressure lines of diagrams from a condensing engine, A 1, represents compression to initial pressure; B 1, compression to 50 pounds, and C, compression to 14 pounds.

The initial pressure of theoretical diagram Fig. 107 is 100 pounds above zero; cut-off one-quarter stroke; terminal pressure 30 pounds above zero, back pressure non-condensing, 15 pounds and condensing 3.5 pounds above zero.

From these diagrams the Available Power may be ascertained, which includes the amount absorbed in driving the engine as well as the effective power, and is usually designated the Indicator Horse Power, and written I. H. P.

This is not the total power derived from the steam, but the I. H. P. must be known in order to ascertain the loss.

The total power diagram of Fig. 107 is shown by the full lines in Fig. 108 and is determined as follows: During a single stroke of the engine piston, the indicator on that side where the steam is admitted will trace the upper line only of the diagram, the lower line being zero.

The pressure of steam during that stroke being measured from the zero line (by the scale of the spring) consequently the total power exerted by the steam is proportional to the area enclosed between the zero line and the upper line of the diagram.

If an indicator be placed in communication with the other side of the piston during the same stroke it will make the lower line only, or the back pressure line, (represented by the dotted lines in Fig. 108) of the indicator diagram. This line also in connection with the zero line forms another diagram, and the area enclosed between these lines, represents the power necessary to expel the exhaust steam during the first part of the stroke, and to compress steam into the clearance spaces during the latter part.

The full lines of the diagram Fig. 108 show the total power exerted by the steam, and the dotted lines show back pressure from exhaust, and compression, or the resistance

which the direct steam must overcome before any useful work can be accomplished.

The difference between the two is the power that must be expended before any is available for running the engine or driving the machinery and is a total loss.

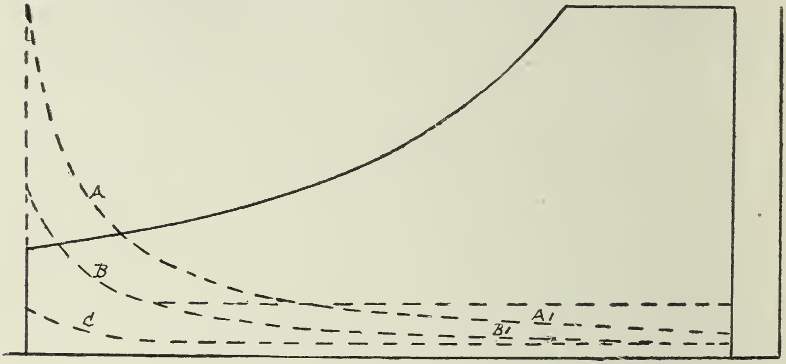


Fig. 108.

It is evident at a glance of the diagram Fig. 108 that this loss is a large proportion of the total power and every means that will tend to reduce this loss should be adopted.

As before mentioned the whole of this cannot be avoided, because it is impossible to obtain a perfect vacuum in an engine cylinder; however the back pressure should be as low as possible, bearing in mind that the clearance spaces should be filled with steam compressed (as near as possible) to the initial pressure at the end of the stroke.

In the case of condensing engines having large clearances, this latter condition is difficult of attainment, and if strictly carried out would probably entail so much loss from increased back pressure as to off-set any gain from high compression.

This points to the benefit of small clearances with which compression may be carried to any desired pressure.

In Fig. 107 the mean effective pressure of the non-condensing diagram compressed to initial pressure, (as line A), is

41 6 pounds, while the average pressure of the total power diagram Fig. 108 is 64 pounds, or about 54 per cent. greater than the available power.

In this case the only way of reducing the back pressure is to commence compression later in the stroke as shown by line B.

The effect of this however, is a loss from low compression, as this is only carried to 50 pounds, and also from increased condensation.

By comparing the results from the two diagrams it will be found that the latter has a mean effective pressure of 47 pounds, making a gain in pressure of about 13 per cent over that of the former but there would be 14 per cent more steam necessary to fill clearance spaces, and the condensation would be increased more than 4 per cent., making a loss of 5 per cent. by not compressing to initial pressure.

If this engine was condensing it would be practically impossible to compress to initial pressure as in order to do so, it would be necessary to begin compression at the commencement of the stroke with a pressure of 6.5 pounds, above zero (line A 1), therefore 8.2 pounds would be the maximum vacuum that could be obtained in the cylinder; consequently the highest pressure to which compression can be carried practically in this case is one-half the initial pressure or 50 pounds above zero (line B 1).

The mean effective pressure of this diagram is 54.3 lbs. and the total power diagram as shown in Fig. 108 is 64 pounds or about 18 per cent. greater; however there would be 14 per cent. more steam required than when compressing to initial pressure, and condensation would be increased fully 4 per cent.

If compression did not commence until later in the stroke and only carried up to 14 pounds (line C), the mean effective pressure would be 59.6 pounds or 9.8 per cent. more than the previous diagram, but over 24 per cent. more steam would be required than if compression were carried to initial pressure,

and the extra condensation would amount to 6.9 per cent. or a loss of 3 per cent. over the previous diagram with greater compression.

Comparing the later diagram (line C), with the total power it will be seen that the total power is but 7.5 per cent. greater, and it is this near approach to the total diagram, that makes the condensing engine so much more economical than the non-condensing, and a moderately well loaded engine gives better results than one lightly loaded, as in the case of a lightly loaded engine, the waste power remains the same, and consequently the percentage of loss increases; although both the total, and available power are less.

In the first diagram the steam developed over 3 horse-power to make 2 horse power available, or a loss of 35 per cent. In some very lightly loaded engines, the steam develops over 3 horse power to make 1 horse power available, or a loss of 70 per cent.

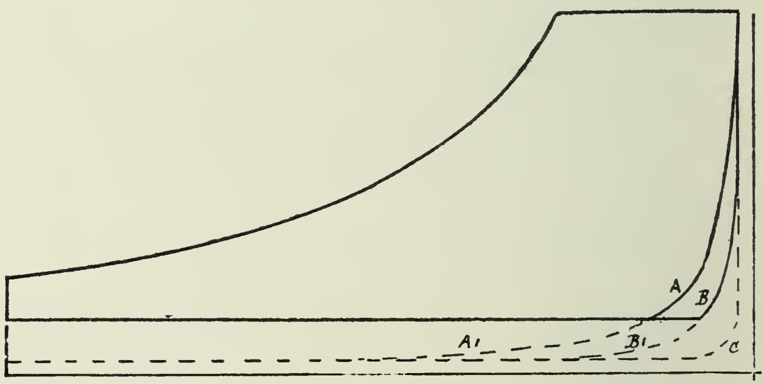


Fig. 109.

THEORETICAL DIAGRAM WITH 2 PER CENT CLEARANCE.

Initial pressure 100 pounds above zero: cut-off, one-quarter stroke; terminal pressure 30 pounds above zero; back pressure non-condensing 15 pounds and condensing 3.5 pounds above zero. In Fig. 109 as in Fig. 107 the full lines represent

diagrams from a non-condensing engine with compression A to initial pressure, and compression B to 50 pounds above zero.

The dotted lines represent the back pressure lines of condensing diagrams, with compression A 1, to initial pressure, also compression B 1, to 50 pounds pressure and compression C to 14 pounds above zero; therefore the two diagrams Figs. 107 and 109 are identical in all respects *except* in their amount of clearance.

The total power diagram of Fig. 109 is shown in full lines in Fig. 110, and the back pressures and the compression curves are shown by the dotted lines A, A 1, B, B 1, and C.

The mean effective pressure being determined in the same manner as in the previous diagrams.

In Fig. 109 the mean effective pressure of the non-condensing diagram compressing to initial pressure (line A) is 44 pounds and the average pressure of the total power diagram Fig. 110 is 60.8 pounds or about 38.5 per cent greater than the available power.

This is quite a loss, but it is a decided improvement over the percentage of loss shown by the diagram with large clearance, as in Fig. 107.

If in this case the compression had been carried to 50 pounds only, (line B), the mean effective pressure would have been 45.6 pounds or 3.6 per cent. greater pressure, but there would be over 4 per cent. more steam required to fill the clearance spaces, and the condensation would be increased over 3 per cent., making a loss of about 3.5 per cent, by the lower compression.

With the condensing diagram Fig. 109 it will be seen that compression may be carried to initial pressure, (line A) and the mean effective pressure in this case would be 52.8 pounds.

If the compression had been carried to 50 pounds only, (line B), the mean effective pressure would have been 56

pounds, or an increase of 6 per cent. in the pressure, but as an off-set, over 4 per cent. *more steam* would be required to fill the clearance space, and the condensation would be increased more than 3 per cent., leaving a loss of 1 per cent.

If the compression had been still lower, (line C), the mean effective pressure would have been 57.5 pounds, or an increase of 8 per cent, in the pressure; but there would be 7 per cent. more steam required to fill the clearances, and condensation would be increased over 5 per cent., making a loss of 4 per cent. by the low compression.

Comparing the results obtained from the two diagrams, the saving effected by an engine having small clearance will readily be seen, and as this has such an important bearing on the economy of the engine, small clearances should be strictly insisted upon.

There are many slow speed engines in use at present having less than 3 per cent. clearance, and this amount should therefore be the maximum allowed in slow speed engines, either simple or compound.

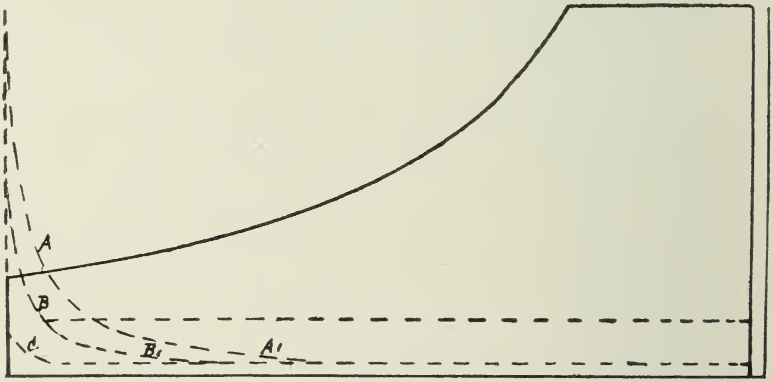


Fig. 110.

Full lines show theoretical diagram of total pressure exerted by the steam.

Dotted lines show back pressure from exhaust and compression, or the resistance which the direct steam must overcome, before any useful work can be accomplished.

COMPARISON OF RESULTS.

Non-Condensing Diagrams.

Clearance	7%	2%	
Cut-Off	25%	25%	
Terminal Pressure	30lbs. above zero	26-5lbs. above zero.	
Back "	15lbs. " "	15lbs. " "	
	M. E. P.	M. E. P.	Gain compared with larger clearance.
Compression to Initial pressure and using equal quantities of steam per stroke in both engines	41·6lbs.	44lbs.	5·8%
Loss by Compressing to 50lbs pressure in place of full Compression	5%	3·5%	7·3%

Condensing Diagrams.

Clearance	7%	2%	
Cut-Off	25%	25%	
Terminal Pressure	30lbs. above zero	26·5lbs. above zero.	
Back "	3·5lbs. " "	3·5lbs. " "	
			Gain compared with larger clearance.
Compression to Initial pressure	—	—	8%
Loss by Compressing to 50lbs. pressure in place of full Compression	6%	1%	13%
Loss by Compression to 14lbs. pressure in place of full Compression	9%	4%	13%

In Compound Engines the greater economy will be obtained when expansion is carried down to the back pressure in each cylinder, except the last, and Compression is carried up to Initial pressure in all; with no drop or free expansion between the cylinders.

CHAPTER XXVIII.

COMBINING DIAGRAMS FROM COMPOUND ENGINES

Compound engines, for almost all purposes are now coming into more general use each year; but in the use of the indicator upon them, both cylinders are treated as simple engines, the power of each being added together.

The diagrams from both cylinders can be taken with the same denomination of spring if desired, but usually a comparatively light spring is used on the low pressure in order that the dimensions or area of the diagram may be increased.

The compound engine with receiver, is as two engines, one high pressure non-condensing, and the other a low pressure condensing engine, but from the fact that the same steam is used in both cylinders, the action of the steam must be considered as if used in a single engine, and the diagrams from each cylinder must be combined, to form an equivalent simple one.

Before making combinations of diagrams from the high and low pressure cylinders of compound engines, the object of combining them should be first understood.

There are certain losses in single or non-compound engines which are corrected to a great extent by compounding, but this in turn introduces other losses which it is desirable to reduce to the least possible amount.

These losses are between the two cylinders, and consist of, condensation in the passages, pipes, and receiver (if one be used), friction in the steam ports and pipes, and expansion of the steam that takes place between the two cylinders without doing useful work.

The extent of these losses can be shown by combining the diagrams from the two cylinders and drawing in the hyperbolic curve. This curve should just touch the expansion line of the high pressure diagram at a point where the exhaust from the cylinder begins, and the space between the curve and both diagrams below this point, and also the space between the two diagrams, represent the loss between the two cylinders.

To correctly combine the two diagrams, the clearance in each cylinder should be known and accounted for, as well as the piston displacement, and the relative length of the two diagrams when combined, is as the ratio of the total volume of the cylinders, that is; the piston displacement plus the clearance at one end.

To do this, a base line may be taken if desired and divided into two parts, which have the same relation to each other in length, as the total volume of the cylinders. The short portion of the line will represent the small or high pressure cylinder, and on this length the diagram from this cylinder is constructed from the lowest pressure, and on the longer portion of the line the diagram from the large cylinder is laid out.

It is best however to decide on the total length of the low pressure diagram first, and a length that can be easily divided into 100 parts will be found most convenient, as percentages of this length can then be easily measured; for example, 10 inches for a scale of tenths, or 12½ inches for a scale of eighths. The combination diagram, Fig. 113, was drawn 12½ inches long, and photo reduced to the length shown.

It now becomes necessary to decide on the scale of the spring to which the two diagrams are to be plotted; usually it

will be found most convenient for this to take the scale of the low pressure diagram; then draw in the atmospheric, and va-

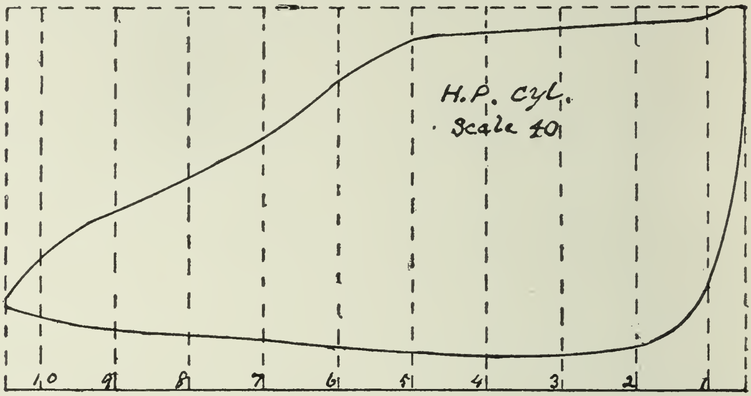


FIG. 111.

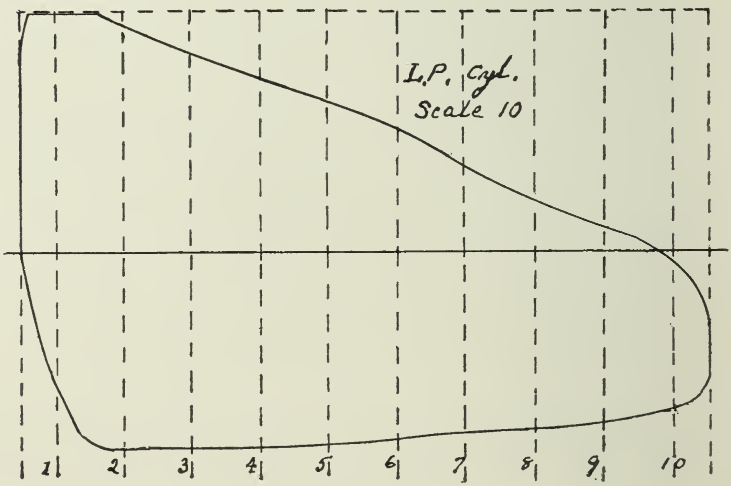


FIG. 112.

cuum lines, and erect perpendiculars at the two extremes of the combination diagram, one of which is the clearance line.

All measurements of distance should be made from the clearance line, and all measurements of pressure from the

atmospheric line. Now divide each original diagram into any desired number of equal parts, 10 being a good number.

Find the volume of the piston displacement of the low pressure cylinder, to which add the volume of the clearance; the total length of the diagram represents this total volume.

Divide the clearance by the total volume, and the quotient will be the percentage this clearance bears to the whole length.

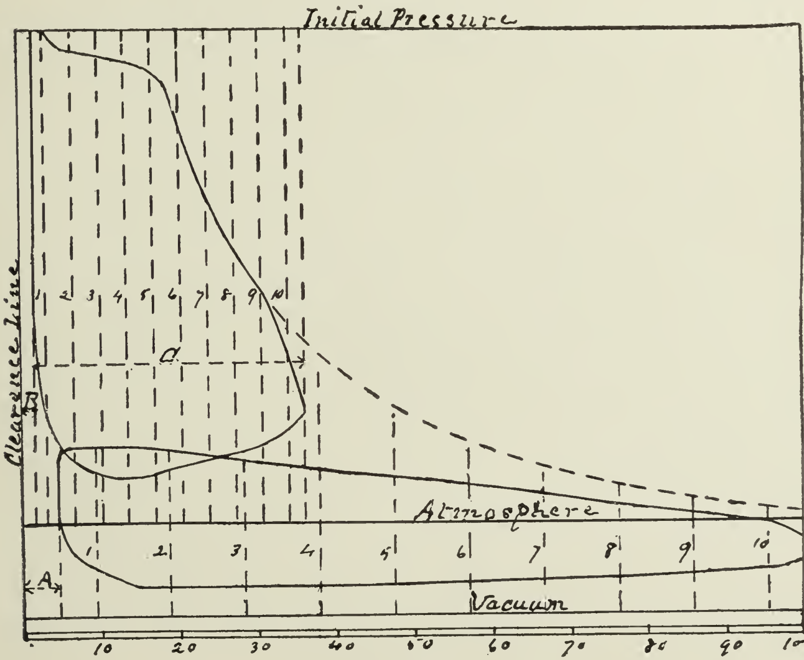


FIG. 113.

Set off this distance from the clearance line, and divide the remainder (representing the piston displacement) into the same number of parts that the original diagram is divided into.

If the scale selected is the same as that of the original diagram, simply transfer the pressures directly with a pair of dividers from the lines on the original diagram to the corresponding lines on the combination; then draw in the connecting

portions of the diagram, and the result will be an elongated diagram from the low pressure cylinder or as if it had been taken with the same spring as before, but with a proportionately enlarged paper drum.

Now find the total volume of the high pressure cylinder, and divide it by the total volume of the low pressure cylinder, and the quotient is the percentage of length of the diagram, which should be measured from the clearance line.

Divide the clearance volume of the high pressure cylinder by the total volume of the low pressure cylinder and measure off the percentage of length, as before, from the line.

Divide the remaining length of the high pressure diagram (representing the piston displacement) into the same number of parts as the original diagram, and transfer the pressures from the lines on the original diagram to the corresponding lines on the combination and to the new scale of pressures.

If the original high pressure diagram was taken with a 40 spring, and the combination diagram made to a scale of 10 lbs. per inch, then the new diagram will be four times as high as before, although it may be shorter.

Next draw in the hyperbolic curve; (a method of doing this is given on page 101, Fig. 56) and the two diagrams thus combined will form a single one.

The process of combining indicator diagrams from compound engines is both interesting, and instructive to the engineer in various ways, and usually attended with most satisfactory results.

Diagram Fig. 113 is a combination of the diagrams Figs. 111 and 112, and were taken from a tandem compound engine.

In order to make the foregoing explanation more clearly understood, diagrams Figs. 111, 112 and 113 have the construction lines shown, and the necessary calculations are given below. Fig. 113 was drawn 12½ inches long over all and reduced by photo engraving process to its present length.

Diameter of H. P. Cylinder - -	30 in.
" " L. P. " - -	50 in.
Stroke of Pistons - - - -	72 in.
Diam. of Piston-rod, both Cylinders	$6\frac{1}{4}$ in.

Volume of High Pressure Cylinder—

$(706{\cdot}86-30{\cdot}68)\times 72=48685$ cubic inches.
Clearance volume = 2545 " "

Total volume 51230

Volume of Low Pressure Cylinder—

$(1963{\cdot}50-30{\cdot}68)\times 72=139163$ cubic inches.
Clearance volume = 7673 " "

Total volume 146836

Total length of L.P. card, $12\frac{1}{2}$ inches, or 100 eights; scale 10. Length of L.P. clearance, marked A on diagram, $\frac{7673\times 100}{146836}$ $=5{\cdot}22\%$ of total length or $\frac{5}{8}+\frac{1}{32}$ in. The remainder of the length, representing the piston displacement, is divided into 10 parts, the same as original diagram, and the pressures transferred to the corresponding lines.

Total length of H.P. card, or B+C on diagram, $\frac{51230\times 100}{146836}$ $=34{\cdot}89\%$, or $4\frac{23}{64}$ in. Length of H.P. clearance, marked B on diagram, $\frac{2545\times 100}{146836}=1{\cdot}73\%$, or $\frac{7}{32}$ in., leaving the distance marked C, representing the piston displacement, which is divided to correspond with the original diagram, and the pressures transferred either with scales or dividers; in the latter case each distance must be multiplied four times. Draw in the connecting portions of the diagrams, taking care to follow the contour of the original as closely as possible; and finally the hyperbolic curve is drawn in.

CHAPTER XXIX.

DIAGRAMS FROM GAS AND OIL ENGINES AND AMMONIA COMPRESSORS.

In the last few years the large increase in the number of gas and oil engines in use for all kinds of manufacturing enterprises both at home and abroad, has been most remarkable, and their number and power are still increasing each year, so that now these motors are in competition with steam engines in almost all progressive countries.

The fact that both gas and oil engines now run with greater regularity than in the past is principally due to improved and better governing arrangements. The portability of small oil engines renders them very convenient for use in country towns, and other places where gas is not made. A greater part of these motors work with the four-cycle, and with lift valves.

Gas engines are in most cases single acting and single cylinder, except for the largest powers, when two cylinders are generally used.

The charge is usually fired either by tube ignition, or by an electric current. The piston speed usually varies from 500 to 700 feet per minute, and the clearance volumes of the cylinder are much larger than in steam engines, usually from 20 to 50 per cent. of the piston displacement, against from 3 to 8 per cent. in steam engines.

When considering that in the employment of gas engines no fuel is being consumed when the engine is not in actual operation, it is evident that they form economical motors when small powers are required, and will soon come into more extensive use as affording a cheap, and efficient motive power in a great number of places where the use of steam is difficult or impossible.

Owing to the greatly increased initial pressure in the cylinders of these engines, (being principally due to the explosive mixture employed therein) specially designed indicators have been constructed to better meet the requirements necessary, and provide means for indicating pressures ranging from 300 to 600 or more pounds pressure per square inch.

This is accomplished by making provision in the construction of the instrument, whereby a piston can be used of a smaller size than the piston of one-half square inch in area, as ordinarily used in most makes of indicators. This smaller piston is usually made one-quarter cf a sq. inch in area, or one-half that of the former, and which when in use, results in doubling the readings or amount of pressure, as when used with the same denomination of spring in connection with the larger piston. Fig. 114 represents the manner of construction in combining the half and quarter inch area piston in one instrument, as applied to the Tabor Indicator, whereby the use of the one-half inch piston for low pressures, or in which the quarter inch piston may be used for the very high pressures often attained in some oil and gas engines. Either of these pistons may be used

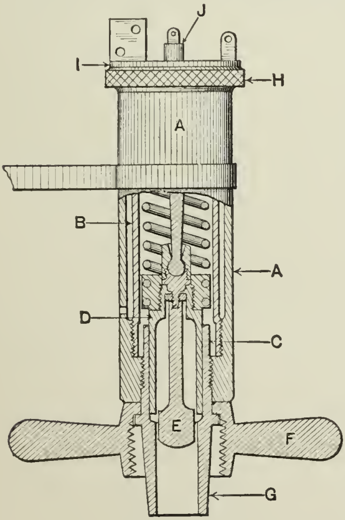


Fig. 114.

independently as desired, without any change whatever either in the spring or any part of the instrument.

In the illustration the half inch piston is not shown; but instead the quarter inch piston is represented attached to the spring in position for operation.

The body A. of the instrument is shown partly in section in order that the location of the parts may be readily observed. B. is the piston cylinder in which the half inch piston works, and which is srcewed at its lower end into the body A. C. is the piston cylinder in which the quarter inch piston works, and is formed in the upper end of the tube G.

D. is the quarter inch piston, and is sufficiently elongated as to reach and work in the cylinder C. Its upper end is threaded and screws into the mounting of the indicator spring. It is made in the form of a shell, and the pencil mechanism is secured to it by means of the extended thumb-nut E. F. is the usual connection for securing the indicator to the cock. H., I. and J. are respectively the cylinder cap, swivel plate and piston rod.

In addition to obtaining the most accurate results from high pressures by the use of the smaller size of piston, this combined indicator has another advantage in that it requires a less number of springs for any given range of pressure.

For example: A 50 spring may be used in connection with the larger piston to 120 pounds pressure per square inch, but by substituting the smaller piston, pressures may be indicated to 240 pounds with the same denomination of spring; a range that would otherwise require two springs; thereby doubling the range of the instrument with a single spring. Figs. 115 to 118 inclusive represent diagrams taken from the Springfield Gas and Gasolene engines. Fig. 115 is from a 7½ inch diameter of cylinder by 14 inches stroke, running 200 revolutions per minute, with scale of spring 120.

Fig. 116 is from a 13 inch diameter of cylinder by 24 inches stroke, running 160 revolutions per minute, with the scale of spring 120.

Fig. 115.

Each of the above may be considered as ideal cards in every respect. Figs. 117 and 118 are from gasolene engines

Fig. 116.

presented simply for the purpose of showing some bad cards. Fig. 117, shows the effect of late ignition, while Fig. 118,

shows also late ignition, and a stratified condition of the charge. This is indicated by the waving expansion line, each waving being an independent explosion or combustion.

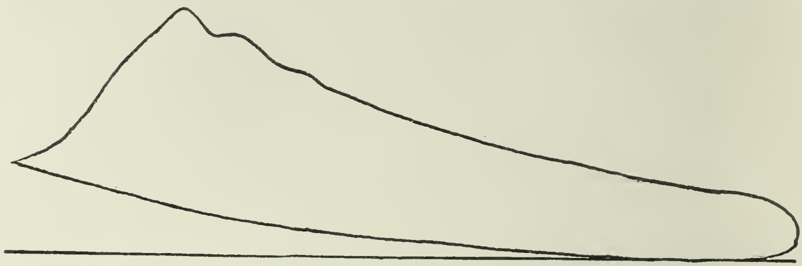


Fig. 117.

These two latter cards were taken expressly for the purpose of showing these very things. Fig. 119 represents a pair of superimposed indicator diagrams taken from a 20 by 26 inch steam cylinder, driving a double-acting ammonia compressor from a Buffalo Refrigerating plant, making 26 revolutions per minute, and are ideal diagrams in almost every respect. However where the speed of the engine is slow, as in refrigerating machines they are only what might be expected, as the exist-

Fig. 118.

ing conditions are generally favorable for the production of good diagrams; because a longer interval of time is given to the steam to pass through the steam ports and fill the cylinder

nearly to boiler pressure at the commencement of the stroke, and thus continue to the point of cut-off; and where the steam admission is regulated by an automatic system of valve gears (such as in this case) we find the cut-off well defined, the ex-

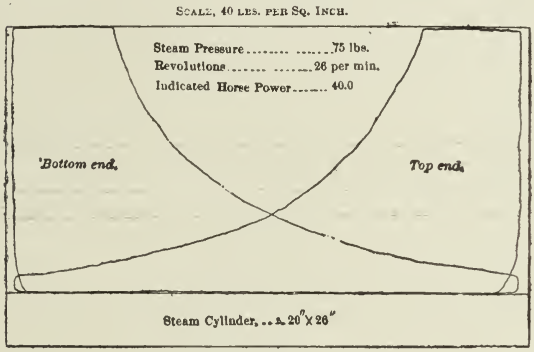


Fig. 119.

pansion line all that could be desired, and the back pressure line nearly straight.

Figs. 120 and 121 represents diagrams taken from the gas or ammonia cylinder which is 15 inches in diameter by 26 inch

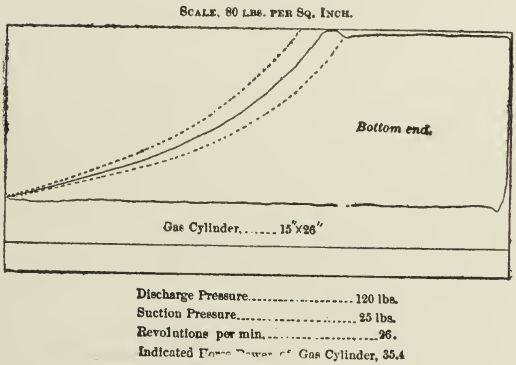


Fig. 120.

stroke. The diagrams from each show that 40 horse power was developed by the steam engine, while 35.4 horse power were needed to compress and discharge the gas within the

compressor. The difference between the two, namely 4.6 horse power, represents the loss caused by friction, amounting to

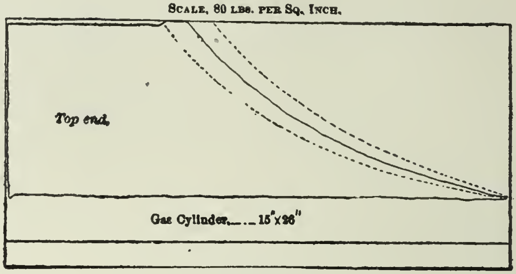


FIG. 121.

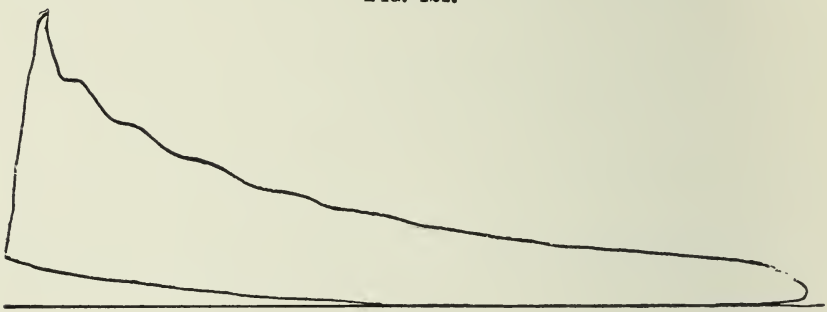


FIG. 122.

11.5 per cent. of the steam engine work, or 13 per cent. of the work accomplished with the gas cylinder.

FIG. 123.

The mean effective pressure of the compressor diagrams Figs. 120 and 121 are computed in the same manner as that for

the steam cylinder and the power developed is determined by the same rule. As the difference is but 4.6 horse power, it demonstrates that 88.5 per cent. of it is made to do useful work.

Fig. 124.

Figs. 122 to 125 represent diagrams from an Otto gas engine with a diameter of cylinder of 6¾ inches, and the length of stroke 15½ inches. The number of explosions per minute being 130, and the number of revolutions per minute 260. Scale of the spring 208.

Fig. 125.

The following original diagrams Figs. 126, 127 and 128 were taken from a rated 7 actual horse power Priestman safety oil engine. Diameter of cylinder 8 inches by 8 inches stroke making 350 revolutions per minute, the scale of the spring

being 100. These three cards comprise a set, being full load, half load, and friction load with the above data.

Fig. 126.

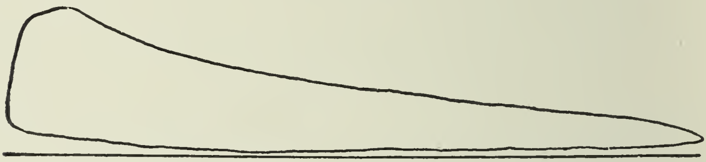


Fig. 127.

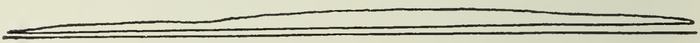


Fig. 128.

A computation of the diagram at full load Fig. 126 shows the engine to be developing 9.41 indicated horse power.

There are still many places where power cannot be obtained, except through some form of heat engine, and where steam is unsuitable, and gas perhaps not available, the oil engine has frequently presented itself and proved quite satisfactory in all such cases.

It is customary to consider oil and gas engines as being practically alike; but there is one notable difference, in that, if an oil engine is run without load it is very liable to stop. This applies only to that type which has no external ignition to itself, neither by electricity nor otherwise, but ignites only by means of a heated chamber which is kept to ignition temperature by the repeated explosions of the charges.

In this class of oil engine the oil is injected by a small pump into what is called the vaporizer; the Otto cycle is worked and ignition takes place when the mixed change of air and vapor becomes compressed in the hot vaporizer, the temperature of which is kept up to redness.

In starting these engines the vaporizer is first heated by a lamp blown by a fan, or by a retort blown by its own self-generated oil gas. After being once heated the vaporizer is kept hot by the recurring explosions. With a light load there are necessarily many explosive strokes cut out by the governor, the same as in case of the gas engine. This so reduces the generation of heat in the cylinder, that the vaporizer is not maintained hot enough to ignite the vaporized oil and the motive power is not produced, and consequently the engine stops. From this it is clear that the size of an oil engine must correspond fairly close with the load to be driven, or else the number of idle strokes will be such, as to prevent the maintainance of a sufficient temperature in the vaporizer to ignite the charge.

CHAPTER XXX.

MAKING CALORIMETER TESTS.

When testing engines to determine their economy, careful tests should be made of the quality of the steam entering the engine, as in many cases water is carried over from the boilers, and condensation in the steam pipes also adds to the amount.

The *priming of boilers* is a serious loss in many steam plants, and for the lack of proper appliances for determining the same, it goes on unchecked, and often unknown.

A large proportion of the loss from the priming of boilers can be prevented if proper precautions be adopted, and thereby a saving of coal effected.

Long steam pipes also invariably cause condensation, and it is very essential that none of this water should pass into the engine, as it occasions a serious loss by promoting initial condensation in the cylinder; therefore, an efficient water separator should be placed in the steam pipe near the engine in order to remove the water of condensation (as near as possible,) from the steam.

The Calorimeters that are being used at the present time, are of three kinds, namely: The condensing or barrel calorimeter, the throttling or superheating, and the wire-drawing.

In the first of these, a certain weight of cold water is utilized to condense a certain weight of steam, and its temperature is raised by the heat in the steam to a certain higher

temperature, depending upon the amount of moisture in the sample of steam condensed. With a device of this kind, (a primitive form,) the appliances for determining the weight, and the temperature must be very sensitive, and the readings carefully observed in order that the results may be approximately accurate.

In the second mentioned, (the throttling calorimeter,) the quantity of heat is ascertained, that is requisite to evaporate, and also slightly superheat the moisture contained in the samply of steam tested, and depends upon the fact, that steam which contains a moderate amount of moisture will become superheated if the pressure is reduced by throttling, without loss of heat.

This instrument is not only easier to use, when the amount of moisture is not excessive, but the quantities are more accurately measured than in some other forms of calorimeters.

It is, however, somewhat limited in its range, and the calculations rather complicated for other than an expert.

The third form of instrument, or wire-drawing calorimeter, operates also by superheating, but neither does this instrument depend on any exterior source of heat for this purpose, because (as in the former case,) steam containing a small amount of moisture when wire-drawn, becomes superheated at the lower pressure; the amount of the superheat depending directly on the percentage of moisture in the steam previous to wire-drawing.

In this instrument the percentage of moisture may be very accurately determined, but like the throttling calorimeter, its range is also somewhat limited, and the calculations complicated for the ordinary engineer.

There is still another form of the instrument, which consists of a wire-drawing device and separator combined.

The steam first enters the water separator, in which nearly all of the moisture is separated from the steam before it passes to the wire-drawing device.

The water which has been separated from the steam may be drawn off, and weighed separately, and the remainder of the moisture is determined by the amount of superheat in the steam after being wire-drawn.

In all of the above mentioned calorimeters, the steam supply is taken from the main steam pipe, by means of a small pipe which extends nearly across the main pipe, and is perforated its entire length in order to obtain as nearly as possible, an average sample of the steam passing through the pipe.

The rules, formulas, and directions for the use of some of the modern forms of calorimeters are commendable for their accuracy and in some cases simplicity, but the principles upon which their operations are based, and results obtained, are in many cases beyond the comprehension or understanding of the ordinary engineer unless he is proficient in the higher branches of mathematics.

Probably no single operation pertaining to the science of steam engineering requires greater care, manipulation, and accuracy in making steam tests than in the use of the calorimeter. Therefore, it will be the endeavor here to present this important subject in as simple a manner as possible in order that engineers not sufficiently versed in the higher mathematics may obtain the benefit of the sciencc connected therewith.

In testing a steam boiler for evaporation alone, without the calorimeter test, but little information is gained so far as the efficiency of the boiler is concerned; because to determine the real efficiency and economy in a steam boiler, the *quality* of the steam generated must be ascertained; as well as the quantity of water evaporated per pound of coal, or per pound of combustible.

The principle cause of priming in boilers is due to a faulty construction, and without the calorimeter test the faulty constructed boiler may, if judged from the amount of water evaporated, show greater efficiency than the properly

constructed boiler. If a boiler on being tested carried off a considerable amount of water with the steam, it would be unfair to credit it with the evaporation of such water into steam, because it has only supplied enough heat to the water carried over to the engine as to raise it from the temperature of the feed water to the boiling point: this quantity of heat being only a fraction of that required for its evaporation: Therefore, every pound of surplus water carried off with the steam takes a correspondingly amount of heat from the boiler, according to the quantity of surplus steam carried over, and without producing an equivalent in work performed. The result of priming is also greatly detrimental to the performance of the engine supplied with steam from such a boiler. It, therefore, becomes necessary with work in which there is to be any degree of accuracy, to determine the quality of the steam used, that is: what percentage of it is actually steam, and how much of it is water.

The main object is to obtain steam as dry as possible, without superheating it; and the boiler that will furnish such steam, and perform the greatest amount in proportion to the amount of fuel consumed, may usually be considered the best boiler, all other things being equal; such as proper setting of boilers, and properly constructed furnaces, etc.

Of the devices here mentioned, the most available, simple, and comprehensive for the working engineer, is the condensing or barrel calorimeter of the primitive form, with which almost any one interested in the subject may equip himself at little expense, and gain a great amount of information not easily attainable by any other means.

It consists of a simple barrel placed upon a platform scale as shown in Fig. 129. In the figure A. represents the main steam pipe, and shows how the attachment is made, and B. represents a standard (made suitable for the purpose,) to which the calorimeter pipe is secured.

That part of the pipe that is inside the main steam pipe should be of one-half inch gas pipe, closed at the end, and perforated, as shown, with small holes about one-eighth inch in diameter.

To the valve attached to the opposite end of this pipe it is a good plan to have a petcock screwed into the top for the purpose of being opened after a test has been completed, to allow any water that may remain in the pipe between the valve and the barrel, to fall to the level of the water in the barrel. The down pipe leading from the valve should have a small rubber hose attached to the end of the pipe, as shown in the figure, and reach to within a short distance of the bottom of the barrel.

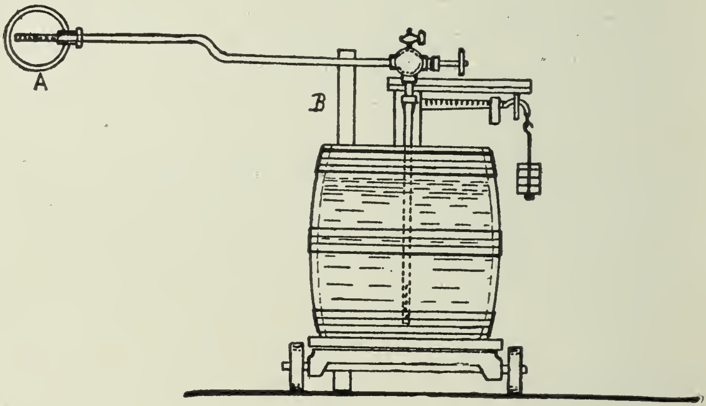


Fig. 129.

The lower end of the hose should be closed, and the hose above that perforated laterally all around with small holes, to avoid the jar due to condensation.

The barrel employed for this purpose should be in such condition as to absorb as little water as possible while making the test.

In platform scales where the beam is graduated to one-half pounds only, it is possible, if so desired, to read to one-tenth or even one-twentieth of a pound, by employing in

connection with the first weight an additional movable weight one-tenth of the weight of that of the first.

There should be suspended in the barrel, (in any convenient manner as will be easily accessible for handling and observation,) an accurately graduated thermometer, capable of being read to at least one-quarter of a degree.

The empty barrel should be accurately weighed and its weight carefully noted; after which (for convenient calculation) put an *even number* of pounds of water into the barrel, leaving sufficient room in the top for the desired amount of condensed steam.

Then set the scales so that they will balance after five or six per cent. more water has been added in the form of condensed steam. Now remove the hose from the barrel, open the valve and let the steam blow through long enough to heat the pipes thoroughly.

During the time the steam is blowing through, take the temperature of the water in the barrel, and carefully make memoranda of the same.

Shut the steam off, insert the hose into the barrel, and again turn the steam on, and as the temperature increases, gently stir the contents with a light wooden stick, in order to insure the mixture being of a uniform temperature at the time the thermometer is read.

It is not advisable to supply steam any longer than to raise the temperature of the water in the barrel to *about* 110 degrees, as beyond this temperature, radiation may take place to such an extent as will complicate the tests, and also the results.

It is desirable to have the water as cold as possible to begin with, as the greater the amount of steam condensed, the less will be a given error in determining its weight and proportion.

In a short time after the steam is turned on, ascertain how near the scale is in balance again, by placing the hand under the scale beam and raising it gently, and as soon as it is found that the scales are about to balance, shut the steam off, open the petcock and then balance the scales.

Then take hold of the thermometer and stir it around gently in the water and carefully observe the highest temperature reached.

Make memoranda of the same, and also note the weight of water in the barrel.

The pipe leading from the main steam should be carefully felted and thoroughly heated previous to each experiment, by wasting steam through it before placing the hose into the calorimeter. Suppose that 360 pounds to be the original weight of water put into the barrel at the beginning of the test; the water being at a temperature of 50 degrees Fahrinheit.

Now suppose steam of 100 pounds guage pressure, (equivalent to 115 pounds absolute pressure) be run into the water until the temperature is increased to 110 degrees Fah., and that upon reweighing its weight has been increased by 20 pounds; this being the amount of steam condensed in raising its temperature.

Referring to the table No. 9, on the properties of water at each degree of *temperature* it will be found that one pound of water at a temperature of 50 degrees Fahrenheit, contains 50.003 heat units; and water at a temperature of 110 degrees Fah. contains 110.110 heat units per pound.

By reference to the table No. 8, on the properties of satuarated steam at different *pressures*, it will be found that steam at 100 pounds gauge pressure, (corresponding to 115 pounds absolute,) contains 1216.97 heat units per pound of steam.

Then we have:

360 pounds = weight of water in the barrel before adding steam.

50.003 = number of heat units per pound of water at 50° Fahrenheit before adding steam.

110.110 = number of heat units per pound of water at 110° Fahrenheit, the temperature after adding steam.

20 pounds = weight of condensed steam and water added to the water in the barrel.

380 pounds = weight of water in the barrel after adding condensed steam and water.

1216.9741 = number of heat units in one pound of dry steam at 100.304 pounds gauge pressure.

The percentage of water in the steam may then be calculated by the following Rule:

First: Subtract the number of pounds of water contained in the barrel before the condensed steam was added from the total number of pounds in the barrel after the condensed steam was added, and multiply the remainder by the total heat units, as shown by the table, contained in one pound of steam due the pressure per square inch indicated by the steam gauge during the time the test was made, and the product will give the total heat units that would have been contained in the steam that has been discharged in the barrel if the steam had been dry.

Second: Multiply the number of heat units contained in one pound of the heated water in the barrel by the number of pounds of such water, and the product will give the total number of heat units contained in the heated water in the barrel.

Third: Multiply the number of heat units contained in one pound of unheated water in the barrel by the number of pounds of that water, and the product will give the total number of heat units in the unheated water in the barrel.

Fourth: Subtract the total heat units in the unheated water in the barrel from the total heat units contained in the water after being heated, and the remainder will give the

number of heat units that have been added by the steam and water discharged into the barrel from the steam pipe.

Fifth: Subtract the heat units that have been added to the original water in the barrel, from the total heat units that would have been contained in the steam if the steam had been dry, and the remainder will show the difference in heat units between dry steam and the steam discharged in the barrel.

Sixth: Multiply the heat units as shown in table No. 8, contained in one pound of steam due the pressure per square inch as shown by the gauge during the test by the number of pounds of steam and water that have been added to the original water in the barrel and the product will give the total number of heat units that would have been contained in the steam that has been discharged in the barrel if the steam had been dry.

Seventh: Divide the difference in heat units between dry steam, and the steam discharged in the barrel, by the total heat units contained in the dry steam, and multiply the quotient by 100, and the product will give the per cent. of water contained in the steam discharged into the barrel.

This rule may be resolved into a formula from the data given as follows:

Let H.=*total heat of steam* at observed pressure=1216.97 heat units per pound.

Let h 1.=weight of water added by heating with steam=20 pounds.

Let W.=weight of water in the barrel before adding steam=360 pounds.

Let w 1.=weight of water in the barrel after adding steam=380 pounds.

Let t.=total heat of water per pound corresponding to initial temperature of water in the barrel at 50° Fah.=50,003 heat units.

Let T. = total heat of water per pound corresponding to final temperature of water in the barrel at 110° Fah. = 110.110 heat units.

Let E. = heating efficiency of the steam furnished compared with the saturated steam between the same limits of temperature.

Let Q. = quality of steam furnished.

Then $Q. = \dfrac{H \times h\,1 - (T \times w\,1 - t \times w)}{H \times h\,1} \times 100 = 2.04+$ per cent.

Or, by figures as per data,

$$Q = \frac{(1216.97 \times \overline{380 - 360}) - (110.110 \times 380 - \overline{50.003 \times 360)} \times 100}{1216.97 \times 20} =$$

2.04+ per cent of water in the steam discharged in the barrel.

The value of E may be ascertained by the following formula:

$E = \dfrac{W\ (T - t)}{h\,1\,(H - T)} = .9774$ heating efficiency of the steam, or by figures $E = \dfrac{360\ (110.110 - 50.003)}{20\,(1216.97 - 110.110)} = .9774 =$ same result as above in heating efficiency.

If the steam is superheated it will show a greater number of heat units per pound for a given pressure than is contained in the standard steam as shown in table No. 8.

The total heat of steam at any given pressure corresponds to a pressure of 14.7 pounds above the given pressure, or that, as shown by the steam gauge.

This plan of making such a test is given as being the most available and most simple of comprehension to the ordinary working engineer. At the same time the greatest care and vigilance must be observed throughout in order to get fairly accurate results, as an error of one-quarter of a pound in determining the amount of steam condensed, will make a difference in the result of about three per cent. Also an error of one-half a degree in temperature will make a difference in the result of at least one and one-quarter per cent.

The scales must be carefully standardized, and as sensitive as possible and greater accuracy may be attained by securing a pointer to the scale beam, and causing it to coincide at each reading with a set mark, by means of small weights of known value placed upon the platform, the reading being corrected accordingly.

The thermometer also must be accurate and delicate; reading preferably (when possible) to tenths of a degree.

The end of the supply pipe must be secured in the main pipe in such a way as to obtain as nearly as possible an average sample of the steam passing through the main steam pipe.

The contents of the barrel during the test should be stirred (as pre-mentioned) thoroughly, to insure a uniform temperature at the time the thermometer is read.

By exercising good judgment, and paying due attention to all these matters of detail, the results should be within two per cent. of correct.

The use of the barrel calorimeter has in many cases, however, been superseded by the throttling calorimeter, especially where very expert tests are necessary.

An improved form of Separating Calorimeter, designed by Prof. R. E. Carpenter, is illustrated in Fig. 130, which has been in use in the laboratories of Sibley College and some other places for the past several years.

The instrument may be described as follows: It consists of two vessels, one being inside the other; the outer vessel surrounds the interior one in such a manner so as to leave a space between them which serves as a steam jacket; the interior vessel is provided with a water gauge glass 10, and a graduated scale 12. The sample of steam, the quality of which is to be determined, is supplied through the pipe 6, into the upper part of the interior vessel.

The water contained in the steam is projected downward into the cup 14 together with the steam, where the course of

the steam and water is changed through an angle of nearly 180 degrees, which causes the greater weight of water by its inertia to be thrown outward through the meshes in the cup and into the space 3 below in the inner chamber.

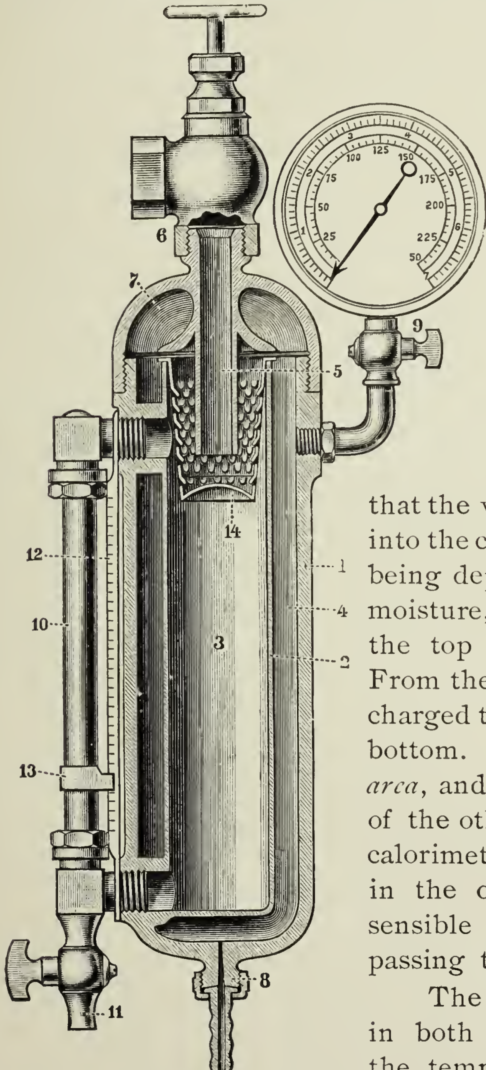


Fig. 130.

The cup serves to prevent the current of steam from taking up any moisture which has already been thrown out by the force of inertia.

The meshes in the cup project upward into the inside of the cup, so that the water intercepted will drip into the chamber 3, while the steam being deprived of a portion of its moisture, passes upward and enters the top of the outside chamber. From the outside chamber it is discharged through an orifice 8, in the bottom. This orifice is *of known area*, and is much smaller than any of the other passages through the calorimeter, consequently the steam in the outer chamber suffers no sensible reduction of pressure by passing through the instrument.

The pressure being the same in both outer and inner chamber the temperature also remains the same; therefore no loss by radiation

can take place from the inner chamber except that which occurs from the exposed surface of the gauge glass. The pressure in the outer chamber and also the flow of steam in a given time is shown by a special graduated gauge attached to the instrument.

The outer circle on the gauge dial is graduated by trial, and shows the weight of steam discharged in ten minutes of time at the observed pressure. The inner circle shows the pressure of steam in the outer chamber.

By a law known as Napiers law, the flow of steam through an orifice from a higher to a lower pressure is in proportion to the absolute steam pressure, until the lower pressure equals or exceeds .6 of that of the higher.

Careful experiments to test the correctness of this law has in all cases indicated its accuracy.

The graduations of the scale 12 attached to the inner chamber show (when the index is properly set), the weight of water in pounds and graduated by hundredths which has been separated from the steam.

This scale is graduated by actual calibration, making it as nearly correct as possible for the temperature of water corresponding to a steam pressure of 100 pounds per square inch.

The percentage of moisture in the steam is found by dividing the weight of water as shown by the water gauge 10, and scale 12, by the sum of this quantity and that shown on the gauge 9. The quality or percentage of dry steam is obtained by dividing the difference of the readings by their sum.

The total size of the instrument is about 10x2½ inches, and its weight about six pounds.

CHAPTER XXXI.

MISCELLANEOUS DIAGRAMS.

The diagrams illustrated in this chapter, some of which represent an excellent distribution of the steam throughout the stroke, while others are quite the reverse, serve to indicate the beneficial results that may be attained by a proper use of the Indicator, and also by its use illustrates the progress made of late years in the distribution of steam in all types of engines. To a skilled engineer the diagram is an index to the steam economy of an engine, as it shows the action of the steam throughout the whole cycle of the pistons movement.

It is a record of the steam from admission to exhaust, and no card can be considered complete which does not indicate every change from the time the steam enters the cylinder, until it is discharged either into the atmosphere, or the condenser.

In treating a card from a compound or multi-cylinder expansion engine, the combined area of all the cards must be considered, according to the scale of each, in order to ascertain the total power developed, and the cost of such power must be obtained from the terminal of the low pressure, or last cylinder, as it is at this point that the useful work of the steam stops; consequently from here the cost must be obtained.

It is an established fact that steam if expanded beyond a certain limit in a *single cylinder* is accompanied by a loss in economy; therefore the only way to increase economy is by more cylinders and greater expansion; hence our present tendency is to higher steam pressures and multi-cylinder engines.

Fig. 131 represents a diagram taken from each end of a Fishkill Corliss Engine. Diameter of cylinder 20 inches by 48

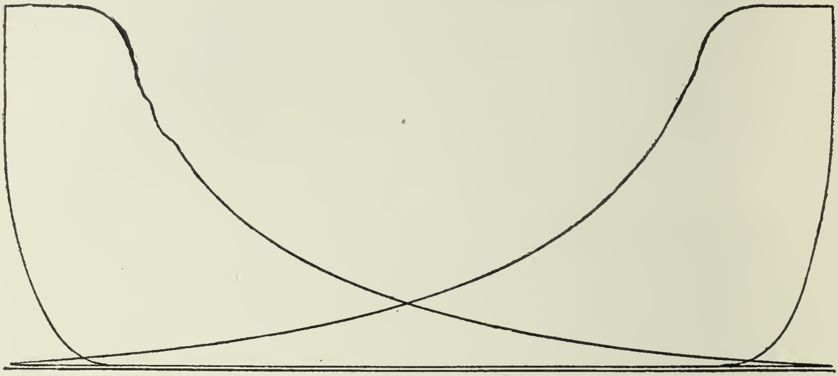


Fig. 131.

inches stroke, revolutions per minute 58. Boiler gauge 84 lbs., scale of spring 40.

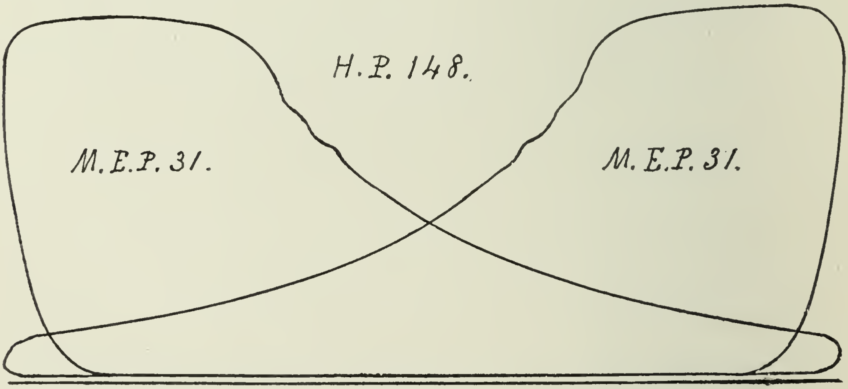


Fig. 132.

Fig. 132 was taken from an automatic slow speed engine. Diameter of cylinder 20 inches, dia. of rod 3 inches, stroke 48 inches making 63 revolutions per minute, 63 lbs. boiler pressure, scale of spring 32.

Figs. 133 and 134 were taken from the same pair of engines as the comparative diagrams represented in Chapter XXIV., the data of the engine being the same as there given.

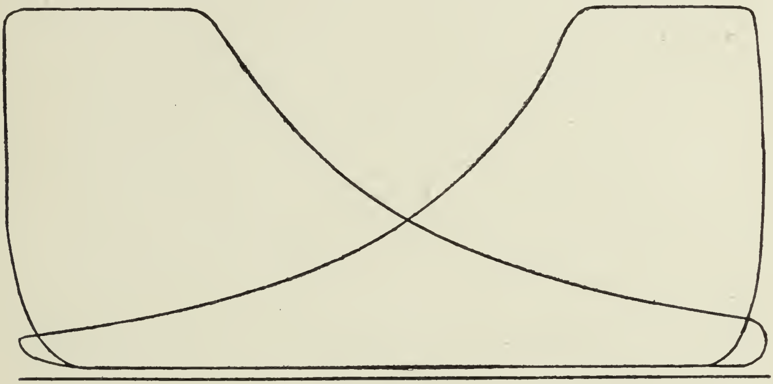


Fig. 133.

In this case the horse power developed being greater than those in the previous chapter.

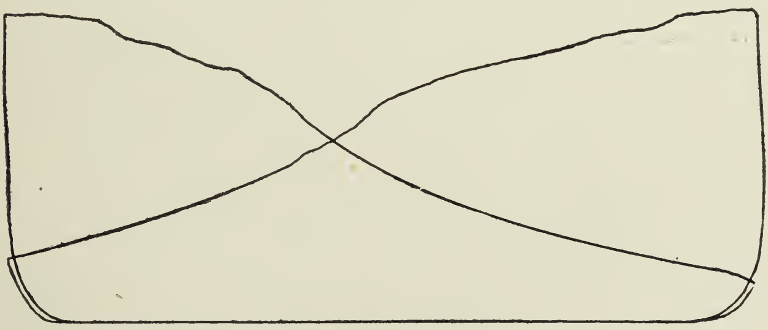


Fig. 134.

The horse power developed by the automatic Fig. 133, being 86.26 while that of the throttling engine, Fig. 134, is 87.06. The mean effective pressure of each is 38 lbs.

Figs. 135 and 136 were taken from a side by side double Wheelock engine, cylinder 18 in. x 48 in. stroke, running 54 revolutions per minute, boiler pressure 90 lbs., the cranks con-

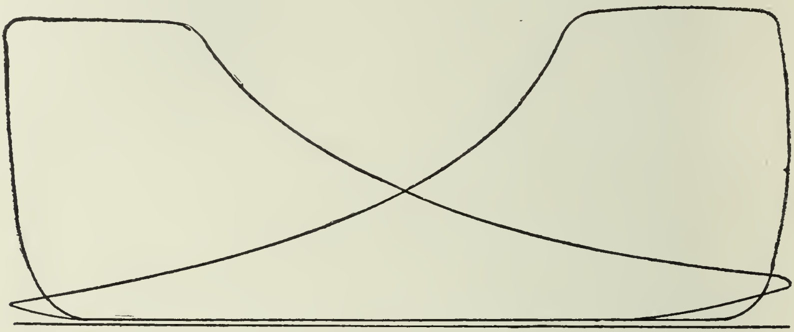


FIG. 135.

nected at right angles or quartering. The M. E. P. of Fig. 135 being 32.2 lbs., and developing 101.75 horse power while the M. E. P. of Fig. 136 is 31.8 and developing 101.44 horse power.

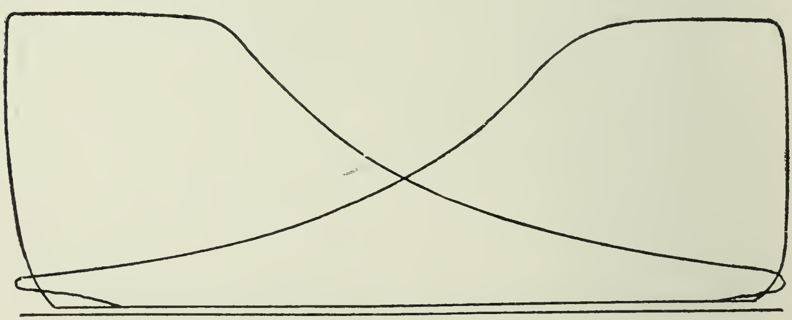


FIG. 136.

Figs. 137 and 138 are diagrams from a Ball & Wood compound, high pressure cylinder 13 inches, and low pressure

cylinder 20½ inches by 15 inch stroke, running 270 revolutions per minute, boiler pressure 145 lbs. Scale of H. P. cyl. 80, and L. P. cyl. 20.

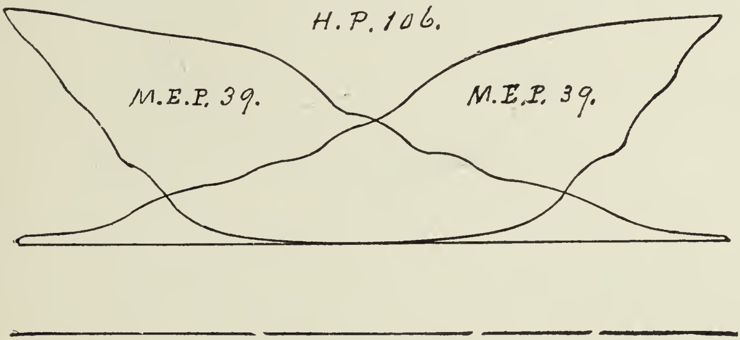


Fig. 137.

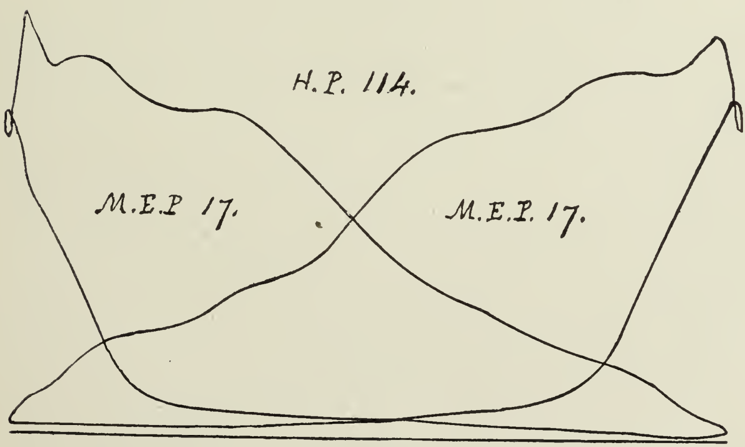


Fig. 138.

Figs. 139 and 140 are diagrams from a 14 in. x 14 in. Fitchburg engine, revolutions 154 per minute. Boiler pressure 75 lbs., by gauge. Scale 60. Fig. 139 shows the condition of an

engine when the indicator was first applied. Fig. 140 shows the improvement made, an increase of about 40 per cent. in

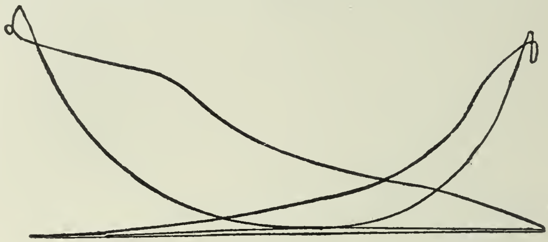


FIG. 139.

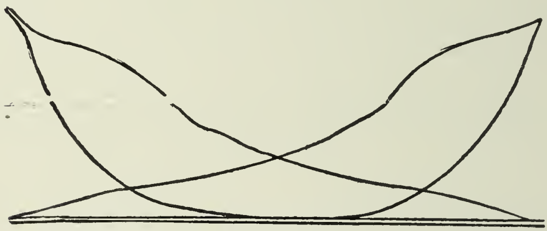


FIG. 140.

mean effective pressure. The cuts are full size, and the load on the engine was 200 incandescent lamps of 16 c. p., and 39 arc lamps, nominally 2000 c. p.

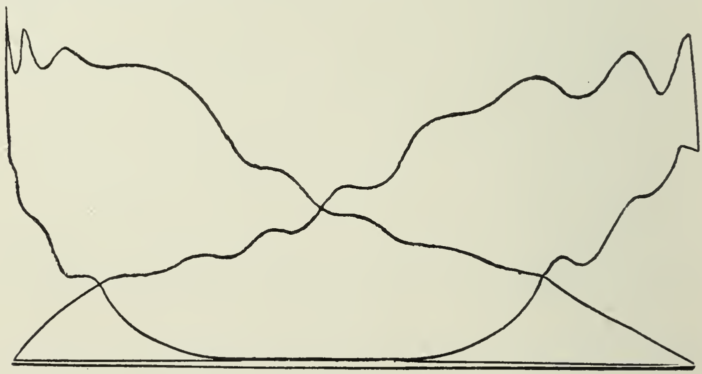


FIG. 141.

Fig. 141 are diagrams taken from both ends of a 13 in. x 13

in. Armington & Sims engine, 250 revolutions per minute. Gauge pressure 85 lbs., and scale 50.

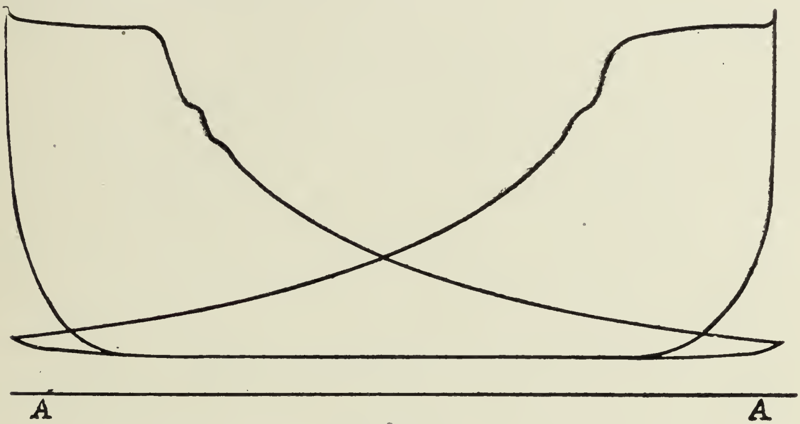


FIG. 142.

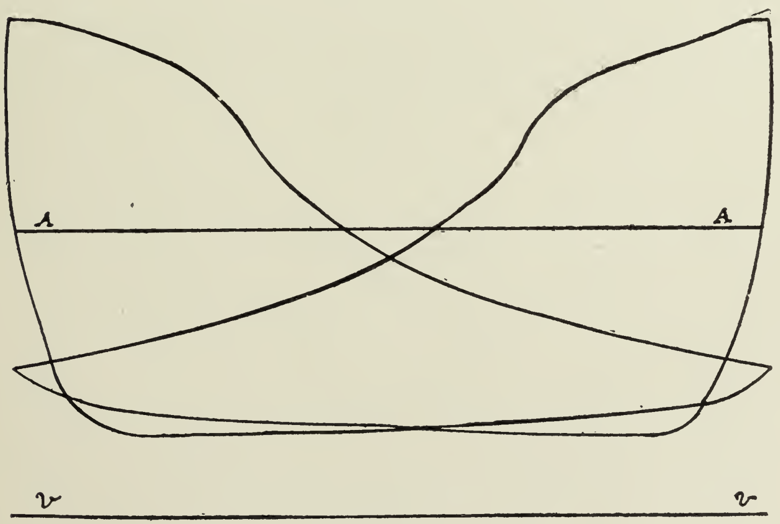


FIG. 143.

Figs. 142 and 143 are diagrams taken from a Watts, Cambell tandem compound H. P. cylinder 18 inches, and low

pressure cylinder 32 inches by 42 inches, stroke running 100 revolutions per minute. Boiler pressure 120 lbs. Scale H. P. 60, and L. P. 10.

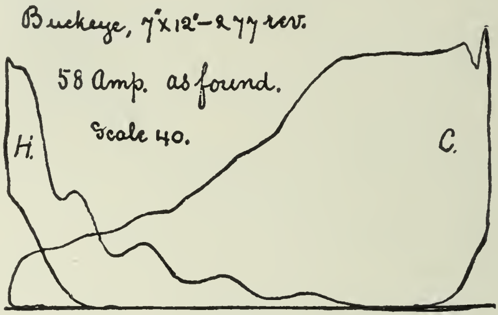


FIG. 144.

Figs. 144 and 145 were taken from a 7 in. x 12 in. Buckeye engine. The former was taken as found running, as per data on card. The latter was taken after the proper equalizing of the valve connections had been made.

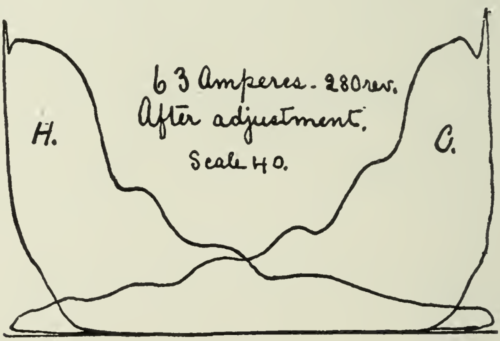


FIG. 145.

Figs. 146 and 147 are diagrams from the high pressure cylinder of a Providence tandem compound engine, taken before and after adjusting. Diameter of cylinder 12 inches.

Stroke 22 inches, revolutions per minute 175. Boiler pressure 120 lbs. Scale 60.

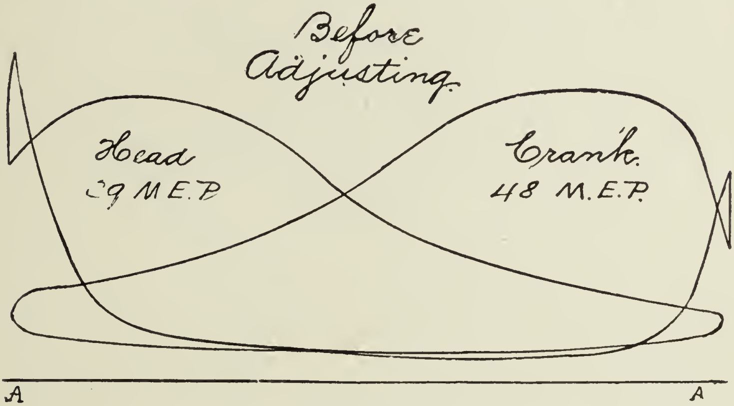


Fig. 146.

The improvement in Fig. 147 consisted in advancing the eccentric on the shaft, and equalizing the valve connections.

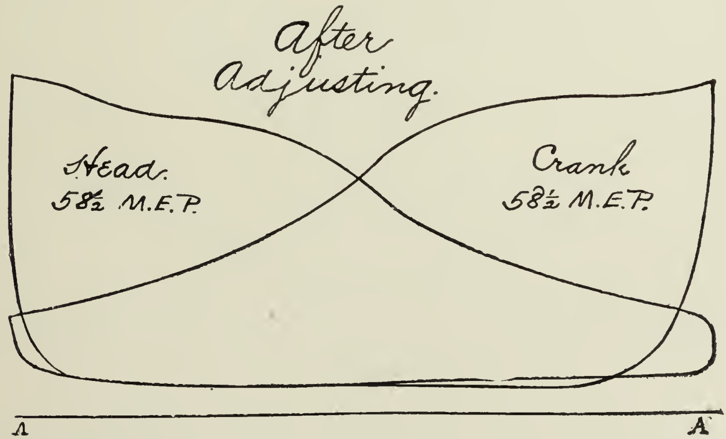


Fig. 147.

Figs. 148 and 149 are diagrams from a Corliss condensing engine, with data affixed thereto. The improvement in Fig.

149 consisted of the same treatment as that given in the two preceeding diagrams.

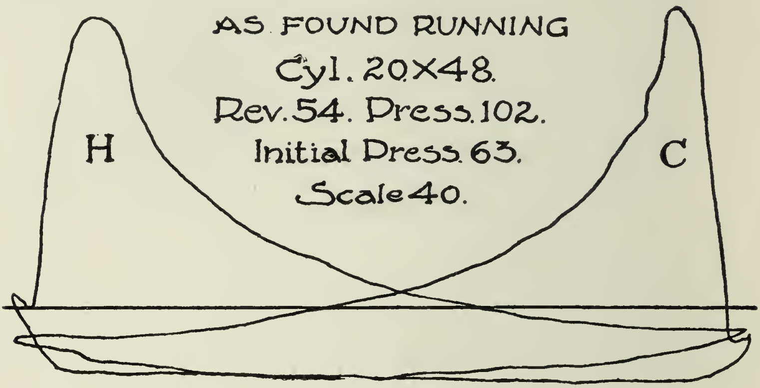


Fig. 148.

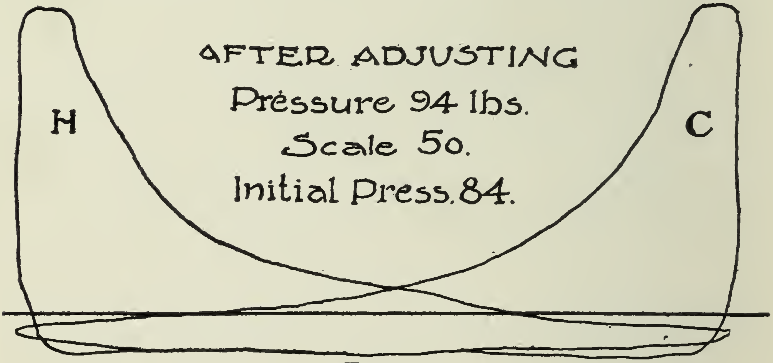


Fig. 149.

The diagrams Fig. 150 were taken from a Porter-Allen condensing engine, 13 inches diameter of cylinder, by 24 inches stroke, 200 revolutions per minute. Boiler pressure 80 lbs. Vacuum 20 inches.

Diagrams, Fig. 151, were taken from a 14 x 24 x 14 inches Westinghouse compound engine, boiler pressure 120 lbs. Scale of Spring 60.

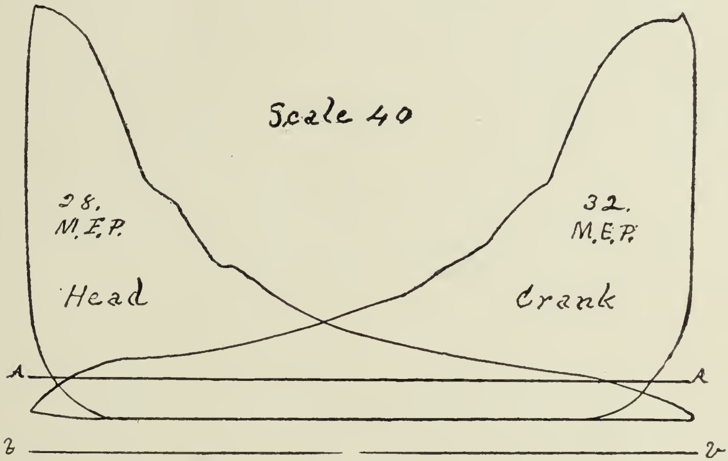


FIG. 150.

Fig. 152 shows a pair of diagrams (Photo reduced in size) taken from a compound tandem jacketed Corliss engine. Diameter of H. Pressure cylinder 16½ inches and L. Pressure

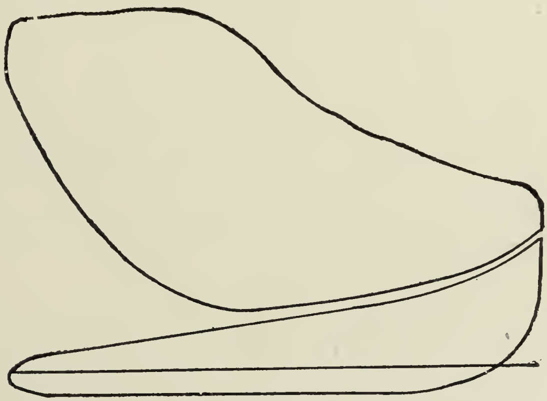


FIG. 151.

cylinder 32 inches; stroke 54 inches, revolutions per minute 59. Boiler pressure 108 lbs.

These diagrams shows the action of the steam while passing through both cylinders, and it will be observed that the steam expanded from an initial pressure of 121 lbs. to 30 lbs. in the first cylinder, with an additional expansion in the second, or low pressure cylinder to 8 lbs., thus giving a range of temperature between 341 deg. and 182 degrees, a change of 159 degrees. It is very evident that any attempt to get the same range of expansion from a single cylinder as obtained in this pair, would be attended with serious loss from condensation; hence, as higher steam pressures are used, and the number of expansions increased, more cylinders are added in order

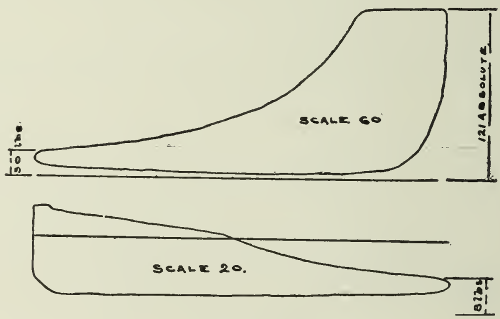


Fig. 152.

to keep the range of temperature in each cylinder within economical limits. Triple and Quadruple expansion engines are simply the results of high steam pressure, and more liberal expansion.

The engines from wnich these diagrams were taken belong to the slow or medium speed type.

In reference to indicator cards in general it will be seen that in many cases their lines do not reach that degree of excellence as shown in Fig. 152.

The fault is often due to bad valve setting or poor valve construction, and it may sometimes be due to the indicator itself, either of which may cause the steam line to be wavy from

start to finish. The usual reason assigned, however, is the presence of water, which comes in such volume that its inertia

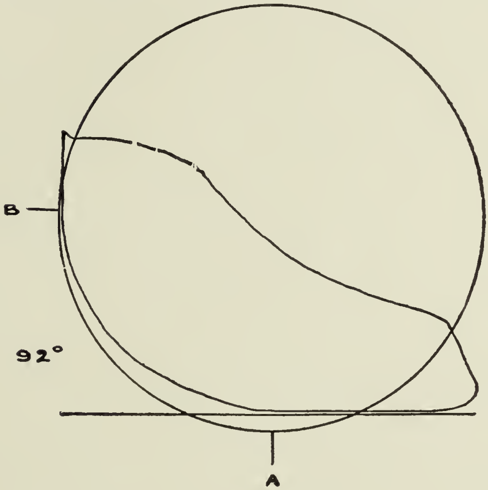


Fig. 153.

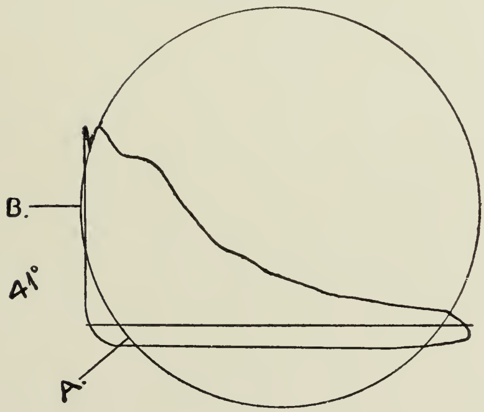


Fig. 154.

carries the indicator piston too far; but the chances are that if water passes in such quantities to the indicator, the engine will

not escape some disaster, and as nothing unusual happens when such cards are taken it is fair to assume some of these irregularities are due to other causes than water; one of which may be considered, and what appear to be the most logical cause.

Diagrams Figs. 153 and 154 were taken from high speed engines, both taken at a speed of 350 revolutions per minute.

The steam and expansion lines on Fig. 153 are all that can be desired; but the lines of Fig. 154 are quite irregular.

On each diagram a circle is drawn to represent the travel of the crank pin. The element of time must be considered, and the influence it has on the indicator piston, spring, and pencil movement. All of the parts have weight, and consequently inertia. If the pencil movement is relatively slow, the inertia, or tendency to go too far, is slight and our diagram will be comparatively free from wave lines: on the contrary where the movement is rapid, or performed in an unusually short time, the inertia will be great and a diagram with irregular lines will be the result, as shown in Fig. 154. The valve motion of an engine influences this time, and in the cases of Figs. 153 and 154 there is enough difference in the valve motion to account for all the difference in the lines of the diagram.

By referring to the diagrams it will be seen in Fig. 153 that the indicator piston begins its upward motion at a point marked A on the exhaust line: at this time the crank is at A on the circle. If the compression line is followed it will be seen that when the indicator pencil arrives at the point B it has reached its limit of upward travel, and the crank has passed on to its point B through 92 deg. of the circle, or more than one-fourth of its entire travel.

Here then is a high-speed engine so far as relative speed is concerned, but an easy speed for indicating; because the large clearance, and early compression makes the movement of the indicator so gradual, that severe inertia shocks are eliminated.

Diagram Fig. 154 is lettered the same but there is a decided difference in the location of the letters. In this case compression did not commence until the stroke was nearly finished, and only rose a few pounds. Ninety per cent. of the upward movement of the indicator pencil is represented by a nearly vertical line, showing that this motion occurred while the crank was passing through a very small part of its travel, that is, from A to B on the circle.

If the difference in the spaces between the points A and B on the two crank circles be compared they will give a fair idea of the difference in the velocities of the pencil movement when these diagrams were taken.

The difference is the measure of the disturbance, and in Fig. 153 will be found all the conditions which insure a smooth card, while Fig. 154 is decidedly the reverse.

In indicator practice we occasionally get cards from slower running engines which show all the irregularities found in cards from high-speed engines: but an analysis of the diagram will probably show that the indicator has had but little help from compression, and the steam admission was very quick.

Most of the excellent diagrams taken from high-speed engines, and published in the catalogues of indicator and engine makers, are usually from compression engines; that is, the type which has large clearance and early compression.

The diagram Fig. 155 is from a pump-cylinder scale 40, and the different lines represent all that can be desired; as the nearest approach to a rectangle in a pump diagram, the better practice it represents.

The line A is the atmospheric line, and the distance from that to the lower line represent the suction, which may be more or less, according to the height the water is lifted, and also to the freedom with which it passes to the pump. The upper line represents the pressure against the plunger or piston necessary to force the water out, and this pressure is due, and

proportionate to the height to which the water is forced, and also to the friction it encounters in passing from the pump.

Commencing at the right hand lower corner of the diagram, (the cylinder being full of water) and the piston begins to move, the pressure instantly rises to about 75 pounds above atmosphere, and continues at a uniform pressure to the end of the stroke, showing that there was no shock due to starting the water-column, and that the passage of the water from the pump-

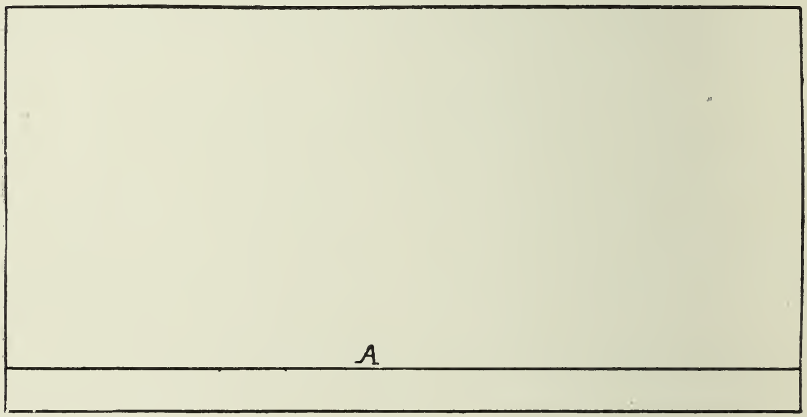


Fig. 155.

cylinder was without additional resistance. If the cylinder is not filled with water, the line at the right will not be vertical.

At the commencement of the return stroke the pressure instantly fell to 8½ pounds below atmosphere, the degree of vacuum necessary to lift the water The lower or suction-line is about as regular as the upper or discharge line showing with what freedom the water passes through the suction-valves. Such a diagram as this shows an absence of shock to the pump, and that a cylinder full of water is discharged at each single stroke.

Fig. 156 is a specimen of diagram which is often taken from pumps, and shows that enormous shocks take place to the parts as well as only partially filling the cylinder with water.

This often happens in practice under circumstances, that cannot always be avoided, but in all cases our endeavors should aim to have the lines of a pump-diagram that will enclose a rectangular figure, and as such, it may be assumed that the operations of the pump must be satisfactory. If the construction of the pump is such that tortuous passages exist, causing undue friction of the water getting into or out of the cylinder

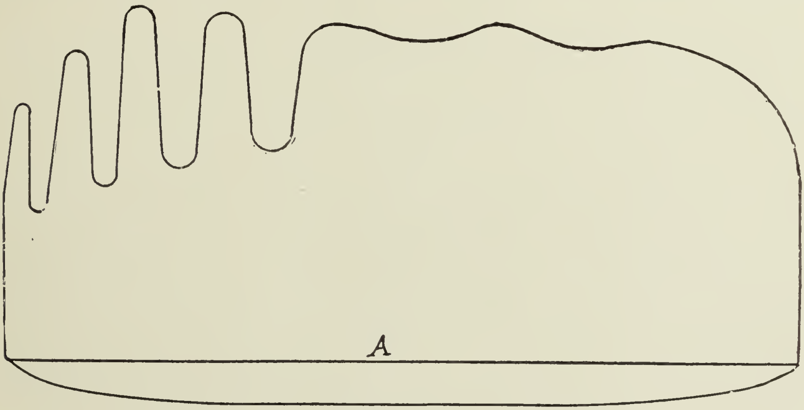


Fig. 156.

the shocks will be greater at some parts of the stroke than at others, and this will be shown by corresponding inclinations of the suction and discharge lines. Shocks and jars and intermittent action will be shown by abrupt irregularities in the lines as in Fig. 156.

Fig. 157. represent diagrams taken from the steam cylinder of a Marsh pump working on a suction lift of 24 feet.

By means of a deflecting valve, the exhausting end of the steam cylinder can, when desired, be placed in open communication with the suction chamber of the pump. The effect of this connection is to extend the vacuum existing in

the suction pipe to the exhaust side of the steam piston. To illustrate the value of this device as claimed for it, is the object of the above card.

The full lines were traced with the steam exhausting directly into the atmosphere. The lever for operating the deflecting valve was then thrown over, thereby turning the exhaust steam into the suction, and the indicator pencil again applied to the same card, thus tracing the dotted outline.

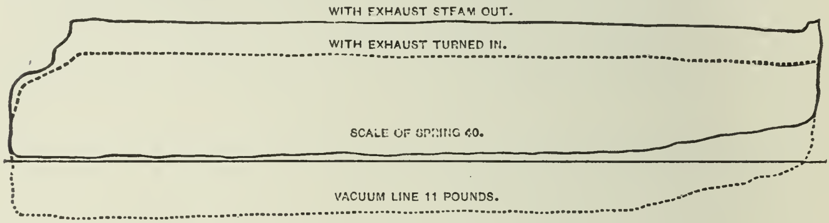


FIG. 157.

From this it is apparent that the steam represented by the area enclosed between the upper full line, and the upper parallel dotted line is just that much gain, for every stroke of the piston, (in this case nearly 25 per cent.), and it will be further noticeable that the total area enclosed by the dotted lines, exceeds the figure enclosed by the full lines to a considerable degree, consequently, there is more power to perform the work, with a smaller expenditure of force, and with the labor a constant factor, the speed of the pump is increased, and a greater amount of water delivered.

CHAPTER XXXII.

ENGINE ECONOMY.

In considering the matter of steam economy in the *engine alone*, it must be understood that the Mean Effective Pressure of the steam acting against the piston for a given time, represents the exact measure or exponent of the work performed by the engine in such time, and is consequently an important factor in all calculations pertaining to engine performance.

The Terminal Pressure, or that pressure of steam which would exist in the cylinder, provided the exhaust valve remained closed to the end of the stroke, is the corresponding measure or exponent of the consumption of steam or water by the engine, *or the cost of the power*, and is also an indispensable factor in the calculation of the diagram.

But almost invariably in all makes of engines, the exhaust valve *opens*, and releases the steam before the piston reaches the end of its stroke; and in such cases the Terminal Pressure is found by continuing the expansion curve in its gradually descending direction (by hand) to the end of the diagram, and measuring from that point to the vacuum line by the scale of the diagram, as shown at T. V., Fig. 77, page 159.

From the conclusions conceded in reference to the Mean Effective and Terminal Pressures, it is evident that the maximum economy will result when the mean effective pressure is greatest relatively to the terminal pressure; therefore if by any means the former can be increased without a corresponding

increase in the latter, or anything that will decrease the latter without correspondingly decreasing the former, must result in improving the economy of the engine.

In non-condensing engines, therefore it would appear that the maximum economy with a given boiler pressure is theoretically obtained when the full pressure is admitted to the cylinder and continued to such point of cut-off, as that the degree of resulting expansion may be such that at the end of the stroke, the terminal pressure has fallen to, or nearly to, atmospheric pressure.

The attainment of this economy in practice will depend somewhat upon conditions, and the construction of the engine; such as possessing a free exhaust for the steam, in combination with the least possible loss from clearance, friction, leakage and condensation.

Hence under favorable conditions it is possible to expand the steam until there is no more work in it, and no greater economy can be expected with a given initial pressure of steam; unless by the aid of a condenser.

With a given load, and boiler pressure, the best theoretical economy is obtained, when the cut-off takes place as early in the stroke, as is consistent with obtaining the average pressure in the cylinder to do the necessary work, and at the same time maintain the required speed of engine.

The measure of the economy of the engine *alone*, therefore is the number of pounds of water which passes through the cylinder in the shape of steam per hour, for each indicated horse power developed.

The actual amount of water thus consumed appears in three conditions; and consists in part of the steam that begins to suffer condensation immediately upon leaving the boiler: due to coming in contact with the comparatively cooler steam passages, and which is further increased upon striking the internal surfaces of the cylinder; part is condensed in the act of

transforming heat into work; that is, in giving motion to the piston, and part in that discharged from the cylinder as exhaust steam.

The portion condensed in the act of changing heat into work is the only one of value; as this quantity (namely, that exhausted and that whose heat is converted into work,) is the amount of water, or steam accounted for by the indicator, and is a measure of the performance of an engine, and when compared with the performance of the best, it shows the economy with which the engine works.

The steam lost in internal condensation is not at all accounted for by the indicator. Hence the total amount of loss from this source is really the difference between the water actually pumped into the boiler, and that accounted for by the indicator.

Tests. It is a very simple matter in testing a plant comprising an engine and boilers, to ascertain the economy of the plant, *as a whole*, as there is usually but little to determine beyond the quantity of fuel consumed, and the horse power developed; but to ascertain the economy and the losses, arising from each of the various parts of a plant, (such as the engines, boilers, heaters, economizers, pipes, etc.), requires close attention to all the several points to insure accurate results.

Where tests of the latter kind are made the following particulars and data should be recorded:

First. The total weight of water supplied to the boiler.

Second. The quantity of water drained from the separator, (if one be used) which includes the water carried along with the steam; (known as priming and for which the boiler alone is responsible) also the condensation in pipes.

Third. The percentage of moisture in the steam that is being supplied to the engine. This may be determined by

means of what is called a calorimeter test, the method of its operation being described in Chapter XXX.

From these amounts the weight of steam (or water) passing through the engine, per hour may be ascertained, and dividing this weight by the horse power developed will give the weight of steam used per horse power per hour.

If this amount is very high it will probably be due to *leakage*, and if such should be the case it will be detected more quickly by this than by any other method.

Fourth. The total weight of coal burned in the furnace.

If this weight is small in comparison with the weight of water pumped into the boiler, showing a large evaporation per pound of coal, it will probably be found that the boiler primes.

If the opposite of this is the case, it may be due to a poor quality of coal, improper firing, poor draft, etc., either of which will cause the final results to be disappointing,

Fifth. The temperature of the feed water before and after passing through the heater; this shows the efficiency of the heater.

In a non-condensing engine the heating of the feed water by the exhaust steam should always be taken advantage of, as in this way a saving of coal will be effected of from 10 to 15 per cent., depending upon the efficiency of the heater and manner of connecting.

To realize the full economy from heating the feed water, it should not enter the boiler at a temperature *less than 210 degrees* Fah. and besides, at this temperature it also obviates the strain on the boiler, that arises from feeding cooler water.

In a condensing engine however there is but little gain from the use of a heater over that of feeding the boiler direct from the hot-well; provided the temperature of the hot-well is not unnecessarily low; *excepting* under circumstances where the water used for condensing purposes is *unfit* for feeding the boiler on account of salt, lime, and other substances held in

solution, and which causes such water to be deleterious in its action upon the interior surface of the boiler.

In the latter case therefore, a slight economy may be derived from the use of a heater; as by its use the fresh water selected for feeding the boiler may have its temperature considerably increased above that of the hot-well while passing through the pipes of the heater on its way to the boiler; and a somewhat further gain is effected, which consists in lessening the amount of water requisite to supply the condenser; due to the heater condensing a portion of the exhaust steam in its passage through it.

In all steam plants there is considerable loss of heat from radiation, by the boiler and setting, and a large percentage of the fuel burned simply replaces the heat radiated from this source, such heat being conveyed away by the air passing over them, without doing any useful work in the way of forming steam; and a further amount is also wasted by the radiation from the pipes, etc., between the boiler and engine, this latter causing the condensation of steam in the pipes.

In order to have a test of this description complete, it is necessary that the amount of these losses from radiation be ascertained, as the heat radiated from the boiler and setting should not be charged against the steam formed; the loss also from condensation in the pipes is an uncertain quantity and often much larger than supposed.

One plan of ascertaining the amount of each of these losses is, after the engine has been stopped, to keep the normal pressure of steam on the boiler for several hours, taking care to keep the water in the boiler, (as near as possible.) at the ordinary level, and the engine stop valve must be tightly closed to prevent any escape of steam or water through it.

The amount of fuel burned, and also the quantity of water pumped into the boiler during this radiation test, should be carefully weighed, and at the end of the test the water of

condensation must be drained from the engine steam pipe, and all other pipes connected directly with the steam space of the boiler, and this total amount also carefully weighed and noted.

If during this test it is found that more water has been supplied to the boiler than that collected from the drains, the difference is evidently due to leakage; therefore, when taking account of the steam passing through the engine in a power test, this leakage should be allowed for.

Hence, to ascertain the exact evaporation per pound of coal, it is necessary that the amount of coal burned, and water used per hour during the *radiation test*, be deducted from the coal and water used per hour during the power test.

The difference in the amount of *coal*, shown by this subtraction, is the actual amount that is consumed in *forming steam only;* while the difference in the amount of *water used*, shows the exact amount of *water* that has been *formed into steam* by this quantity of coal.

Therefore, in power tests where the amount of coal consumed is the measure of the engine's performance, (as is frequently the case,) the quantity of coal remaining (after deducting the amount used in the radiation test,) is the correct amount chargeable against the engine.

In making the radiation test, every precaution should be taken that will tend to burn the coal to the best advantage; the draft openings for the furnace, and also the back damper, should be carefully adjusted, so as to just maintain the pressure of steam required, and also to prevent an excess of air from conveying any large amount of heat up the chimney.

In the absence of an accurate water metre for measuring the quantity of water forced into the boiler during a test, a very satisfactory arrangement may be substituted, consisting of two barrels or casks, in connection with an ordinary platform weighing scales.

One of these barrels is placed upon scales, and together elevated above the second barrel, which for convenience, should be somewhat the larger.

The feed water is drawn into and carefully weighed in the upper barrel, and then run off into the lower one, from which it is pumped into the boiler.

Another and somewhat more convenient method of testing is sometimes resorted to, but which gives approximate results only. In this operation the feed water is brought to a given point near the upper part of the gauge glass, and then shut off, and the test made by observing the rate at which the water boils away.

The height of the water in the glass at the beginning, and at the end of the test being carefully observed and noted.

The weight of the water evaporated and supplied to the engine can then be calculated from the cubical volume that it occupied in the boiler, always bearing in mind that the *weight of a given volume of water* varies with its temperature. (See table No 9).

To insure greater accuracy, tests made by this method can be repeated a number of times, and the results averaged.

Feed water tests, made by measuring all of the water supplied to the boiler, are of no positive value unless leakage of water from the boiler (if any exist) be deducted therefrom; hence, particular attention should always be given to this fact, and the leakage determined as before described.

A better and more accurate way than either of the above methods for ascertaining *the weight of steam consumed by an engine*, is to use a Surface Condenser, in which all of the steam passing through the engine is condensed, and the resulting water saved and weighed; and the only correction needed is to deduct the per cent. of moisture contained in the steam supplied, which may be determined by a Calorimetric test.

A portion of the steam required by an engine may also be found by calculation from the diagram.

A method of making this calculation is given in Chapter XXI.

Engine economy includes everything that enters into cost of maintenance, and operation, and the problem with the engineer in charge of engines and boilers, is how to get the best possible results from such machinery as comes under his direction.

The value of his services depends largely upon his ability in this direction, and an important part of his education is how best to accomplish the most desired and economical results.

Economy to him consists in keeping the fuel account as low as possible for the power developed, having few repairs, little loss (through accidents or otherwise) from stoppages, and also having the least possible loss from wear and tear or deterioration.

The cost of fuel is *always* an important matter, but sometimes it happens of more importance that there be no compulsory stoppage of the engine or that the speed be very regular.

It is the province of the engineer to study this in any particular instance and govern himself according to circumstances and observation, and take measures to obviate or remove (if possible) whatever may be detrimental to good economy.

In reference to fuel economy it frequently happens that the engineer has to contend with defective conditions, or under such adverse circumstances, as will render the attainment of good economy impossible.

A condition very unfavorable to fuel economy of non-condensing engines exists in cases where the expansion line of the indicator diagram falls below the pressure of the atmosphere early in the stroke (as shown in Fig. 62), or in other

words, where the engine is too large for its work, necessitating an early cut-off, and in consequence, greater loss from condensation.

The reason of this increased loss through condensation, is owing to the interior walls of the cylinder becoming cooled to a lower temperature during expansion and exhaust, and in consequence, a considerable portion of the entering steam at the beginning of the stroke is condensed; due to parting with its latent heat in order to restore the temperature of the interior exposed surface of the cylinder.

In an engine with a light load, the steam thus condensed is a larger proportion of the total steam used, than in one more heavily loaded. Another reason why poor economy is generally the rule where light loads prevail, is that a part of the work done in the cylinder of a steam engine is in overcoming the *friction of the moving parts;* and this friction does not increase proportionately fast as the load is increased, the friction sometimes being nearly as great with light running, or no load, as with the engine fairly well loaded.

In a non-condensing engine the useless work of moving the piston against the pressure of the atmosphere must always be done, besides some additional back pressure, although this will not increase as fast in proportion as the mean effective pressure is increased. In a condensing engine the piston has always to be moved against pressure due to imperfect vacuum, besides a certain amount of back pressure also. Owing to various conditions and circumstances, connected with the subject, the *exact loss* from condensation cannot be ascertained very closely by calculation; therefore it cannot be told just what the mean effective pressure on an engine piston *should be*, to realize the best economy in fuel consumption.

Experiments to determine the relation of steam consumption to point of cut-off, under different pressures of steam in a non-condensing engine will be found described in Chapter XXVI.

CHAPTER XXXIII.

The following table (No. 6) apply both to exhaust steam heaters and economizers where, what would otherwise be, waste heat is utilized for heating the feed water.

The percentage of saving given is the saving in the amount of heat required to generate a certain quantity of steam. The saving in fuel depends on other conditions, and may be more than given above. If, for instance, a boiler is too small to steam easily without a feed-water heater, the application of a heater will make a much greater saving in fuel than the percentage given in the table: but if the boiler steams easily without a heater, the addition of a heater will save about the same per cent. of fuel as given in the tables. It is assumed in each case that the addition of an exhaust steam heater does not impair the vacuum on a condensing engine, or increase the back pressure on a non-condensing engine, and that the addition of an economizer does not impede the draught.

A heater may be applied to the exhaust pipe of a condensing engine that will, without impairing the vacuum, heat the feed-water from the temperature of the hot well (about 100°) to 165° or 170°, a saving of about 6 per cent., then passing it through an economizer, should raise the temperature another 100° (from 170° to 270°), making a further saving of about 10 per cent.

In non-condensing engines an exhaust steam heater will heat the feed-water from 62° to 210°, a saving of 12.9 per cent.

An economizer will heat the feed-water to from 220° to 320°, according to the temperature of the waste gases, and also the temperature of the water entering the economizer.

After heating the water, care should be taken that it stays hot until it enters the boiler. If, for instance, the water is heated in an economizer to 270°, and then in passing through the pipes to the boiler it cools down 10°, to 260°, there is a loss of 1.06 per cent., the greater portion of which might be saved by carefully protecting the pipes.

The temperature of the feed water, before and after passing through the economizer, and the temperature of the gases both sides of it, will show whether the economizer is efficient or not. If the temperature of the water leaving the economizer gradually lowers while the average temperature of the escaping gases gradually increases, it indicates a scaling up of the economizer, which at once requires attention. Also if the quantity of fuel increases gradually, it may possibly be due to air leaks in the setting or scaling either in the boiler, or economizer, and should be remedied.

Table No. 6.

Saving effected by the use of Feed-Water Heaters in the generation of steam of 100 lbs. guage pressure or 115 lbs. total pressure.

Temp. from which the water is heated or cooled.	Temperature of the Water entering Boiler.												
	32°	35°	40°	45°	50°	55°	60°	62°	65°	70°	75°	80°	85°
	Percentage of gain (+) or loss (—) by heating or cooling the water.												
		+	+	+	+	+	+	+	+	+	+	+	+
32°	0·0	·25	·67	1·10	1·52	1·94	2·36	2·53	2·79	3·21	3·63	4·05	4·47
35	—·25	·00	·42	·85	1·27	1·69	2·12	2·29	2·54	2·96	3·39	3·81	4·23
40	·67	—·42	·00	·43	·85	1·28	1·70	1·87	2·13	2·55	2·98	3·40	3·83
45	1·10	·85	—·43	·00	·43	·85	1·28	1 45	1·71	2·13	2·56	2·99	3·41
50	1·52	1·27	·85	—·43	·00	·43	·86	1·03	1·29	1·71	2·14	2·57	3·00
55	1·94	1·69	1·28	·85	—·43	·00	·43	·60	·86	1·29	1·72	2·15	2·58
60	2·36	2·12	1·70	1·28	·86	—·43	·00	·17	·43	·86	1·30	1·73	2·16
62	2·53	2·29	1·87	1·45	1·03	·60	—·17	·00	·26	·69	1·13	1·56	1 99
65	2·79	2·54	2·13	1·71	1·29	·86	·43	—·26	·00	·43	·87	1·30	1·74
70	3·21	2·96	2·55	2·13	1·71	1·29	·86	·69	—·43	·00	·44	·87	1·31
75	3·63	3·39	2·98	2·56	2·14	1·72	1 30	1·13	·87	—·44	·00	·44	·88
80	4·05	3·81	3·40	2·99	2·57	2·15	1·73	1·56	1·30	·87	—·44	·00	·44
85	4·47	4·23	3·83	3·41	3·00	2·58	2·16	1·99	1·74	1·31	·88	—·44	·00
90	4·90	4·66	4·26	3·85	3·44	3·02	2·60	2·43	2·18	1·75	1·32	·89	—·45
95	5·33	5·09	4·68	4·28	3·87	3·45	3·04	2·87	2·61	2·19	1·76	1·33	·89
100	5·75	5·51	5·11	4·70	4·30	3·88	3·47	3·30	3·05	2·63	2·20	1·77	1·33
110	6·59	6·36	5·96	5·56	5·15	4·74	4·33	4·17	3·92	3·50	3·07	2·65	2·22
120	7·44	7·20	6·81	6·41	6·01	5·60	5·20	5·03	4·79	4·37	3·95	3·53	3 10
130	8·29	8·06	7·67	7·27	6·88	6·47	6·07	5·91	5·66	5·25	4·34	4·42	4·00
140	9·14	8·80	8·52	8·13	7·73	7·33	6·93	6·77	6·53	6·12	5·71	5·30	4·88
150	9·99	9·76	9·38	8·99	8·60	8·20	7·81	7·65	7·41	7·00	6·60	6·19	5·77
160	10·84	10·61	10·23	9·85	9·46	9·07	8·68	8·52	8·29	7·89	7 48	7·07	6·66
170	11·68	11·46	11·08	10·70	10·32	9·93	9·55	9·39	9·15	8·76	8·36	7·95	7·55
180	12·54	12·31	11·94	11·57	11·19	10·80	10·42	10·26	10·03	9·64	9·24	8·84	8·44
190	13·39	13·17	12·80	12·43	12·05	11·67	11·29	11·14	10·91	10·52	10·13	9·73	9·33
200	14·24	14·02	13·66	13·29	12·92	12·54	12·17	12·01	11·79	11·40	11·01	10·62	10·23
210	15·10	14·89	14·53	14·16	13·79	13·42	13·05	12·90	12·67	12·29	11·91	11 52	11 13
212	15·27	15·06	14·69	14·33	13·96	13·59	13·22	13·07	12·85	12·47	12·08	11·69	11·30
220	15·96	15·74	15·38	15·02	14·66	14·29	13·92	13·77	13·55	13·17	12·79	12·41	12·02
230	16·81	16·60	16·24	15·89	15·52	15·16	14·79	14·65	14·42	14·05	13·68	13 30	12·91
240	17·66	17·45	17·10	16·75	16·39	16·03	15·67	15·52	15·30	14·93	14·56	14·18	13·81
250	18·52	18·32	17·97	17·62	17·26	16·91	16·55	16·41	16·19	15 82	15·45	15·08	14·71
260	19·38	19·18	18 83	18·49	18·14	17·78	17·43	17·29	17·0[illegible]	16·71	16·35	15·98	15·61
270	20·24	20·03	19·69	19·35	19·01	18·66	18·31	18·17	17 95	17 59	17·23	16·87	16·50
280	21·09	20·89	20·55	20·21	19·87	19·52	19·18	19·04	18·83	18·47	18·12	17·76	17·39
290	21·95	21·75	21·42	21·08	20·75	20·40	20·06	19·92	19·71	19·36	19·01	18·65	18·30
300	22·80	22·61	22·28	21·95	21·61	21·27	20·93	20·80	20·59	20 25	19·[illegible]0	19·54	19·19
310	23·67	23·47	23·14	22·82	22·49	22·15	21·82	21·68	21 48	21·14	20·79	20·44	20·09
320	24·52	24·32	24·00	23·68	23·35	23·02	22·69	22 56	22·35	22·02	21·67	21·33	20·98

TABLE NO. 6.—CONTINUED.

Saving effected by the use of Feed-Water Heaters in the generation of steam of 100 lbs. guage pressure or 115 lbs. total pressure.

Temp. from which the water is heated or cooled.	Temperature of the Water entering Boiler.												
	90°	95°	100°	110°	120°	130°	140°	150°	160°	170°	180°	190°	200°
	Percentage of gain (+) or loss (—) by heating or cooling the water.												
	+	+	+	+	+	+	+	+	+	+	+	+	+
32°	4·90	5·33	5·75	6·59	7·44	8·29	9·14	9·99	10·84	11·68	12·54	13·39	14·24
35	4·66	5·09	5·51	6·36	7·20	8·06	8·80	9·76	10·61	11·46	12·31	13·17	14·02
40	4·26	4·68	5·11	5·96	6·81	7·67	8·52	9·38	10·23	11·08	11·94	12·80	13·66
45	3·85	4·28	4·70	5·56	6·41	7·27	8·13	8·99	9·85	10·70	11·57	12·43	13·29
50	3·44	3·87	4·30	5·15	6·01	6·88	7·73	8·60	9·46	10·32	11·19	12·05	12·92
55	3·02	3·45	3·88	4·74	5·60	6·47	7·33	8·20	9·07	9·93	10·80	11·67	12·54
60	2·60	3·04	3·47	4·33	5·20	6·07	6·93	7·81	8·68	9·55	10·42	11·29	12·17
62	2·43	2·87	3·30	4·17	5·03	5·91	6·77	7·65	8·52	9·39	10·26	11·14	12·01
65	2·18	2·61	3·05	3·92	4·79	5·66	6·53	7·41	8·29	9·15	10·03	10·91	11·79
70	1·75	2·19	2·63	3·50	4·37	5·25	6·12	7·00	7·89	8·76	9·64	10·52	11·40
75	1·32	1·76	2·20	3·07	3·95	4·84	5·71	6·60	7·48	8·36	9·24	10·13	11·01
80	·89	1·33	1·77	2·65	3·53	4·42	5·30	6·19	7·07	7·95	8·84	9·73	10·62
85	·45	·89	1·33	2·22	3·10	4·00	4·88	5·77	6·66	7·55	8·44	9·33	10·23
90	·00	·44	·89	1·78	2·66	3·56	4·45	5·34	6·24	7·13	8·02	8·92	9·82
95	—·44	·00	·45	1·34	2·23	3·13	4·02	4·92	5·82	6·71	7·62	8·52	9·42
100	·89	—·45	·00	·90	1·79	2·70	3·59	4·50	5·40	6·30	7·20	8·11	9·01
110	1·78	1·34	—·90	·00	·90	1·82	2·72	3·63	4·55	5·45	6·36	7·28	8·19
120	2·66	2·23	1·79	—·90	·00	·92	1·83	2·75	3·68	4·59	5·51	6·43	7·35
130	3·56	3·13	2·70	1·82	—·92	·00	·92	1·85	2·78	3·70	4·63	5·56	6·49
140	4·45	4·02	3·59	2·72	1·83	—·92	·00	·94	1·88	2·81	3·74	4·68	5·62
150	5·34	4·92	4·50	3·63	2·75	1·85	—·94	·00	·95	1·89	2·83	3·78	4·73
160	6·24	5·82	5·40	4·55	3·68	2·78	1·88	—·95	·00	·95	1·90	2·86	3·82
170	7·13	6·71	6·30	5·45	4·59	3·70	2·81	1·89	—·95	·00	·97	1·93	2·90
180	8·02	7·62	7·20	6·36	5·51	4·63	3·74	2·83	1·90	—·97	·00	·97	1·95
190	8·92	8·52	8·11	7·28	6·43	5·56	4·68	3·78	2·86	1·93	—·97	·00	98
200	9·82	9·42	9·01	8·19	7·35	6·49	5·62	4·73	3·82	2·90	1·95	—·98	·00
210	10·72	10·32	9·93	9·11	8·28	7·43	6·57	5·68	4·78	3·87	2·93	1·98	—1·00
212	10·90	10·51	10·10	9·29	8·46	7·61	6·75	5·87	4·97	4·06	3·13	2·17	1·20
220	11·62	11·23	10·83	10·02	9·20	8·36	7·51	6·63	5·74	4·84	3·91	2·95	2·00
230	12·52	12·13	11·74	10·94	10·12	9·29	8·44	7·58	6·69	5·80	4·88	3·95	2·99
240	13·42	13·03	12·64	11·85	11·04	10·22	9·38	8·52	7·65	6·77	5·86	4·93	3·99
250	14·32	13·94	13·55	12·77	11·97	11·16	10·33	9·48	8·62	7·74	6·84	5·93	4·99
260	15·22	14·85	14·47	13·69	12·91	12·10	11·28	10·44	9·58	8·72	7·83	6·92	6·00
270	16·12	15·75	15·37	14·61	13·83	13·03	12·22	11·39	10·54	9·68	8·80	7·90	6·99
280	17·02	16·65	16·28	15·52	14·75	13·96	13·16	12·33	11·50	10·65	9·78	8·89	7·98
290	17·92	17·56	17·19	16·44	15·68	14·90	14·10	13·29	12·46	11·62	10·76	9·88	8·99
300	18·82	18·46	18·09	17·36	16·60	15·82	15·04	14·24	13·42	12·59	11·74	10·87	9·98
310	19·73	19·37	19·01	18·28	17·53	16·76	15·99	15·19	14·38	13·57	12·72	11·86	10·99
320	20·62	20·27	19·91	19·19	18·45	17·69	16·93	16·14	15·34	14·53	13·70	12·85	11·98

Table No. 6.—Continued.

Saving effected by the use of Feed-Water Heaters in the generation of steam of 100 lbs. guage pressure or 115 lbs. total pressure.

Temp. from which the water is heated or cooled.	Temperature of the Water entering Boiler.												
	210°	212°	220°	230°	240°	250°	260°	270°	280°	290°	300°	310°	320°
	Percentage of gain (+) or loss (—) by heating or cooling the water.												
	+	+	+	+	+	+	+	+	+	+	+	+	+
32°	15 10	15·27	15·96	16·81	17·66	18·52	19·38	20·24	21·09	21·95	22·80	23·67	24·52
35	14·89	15·06	15·74	16·60	17·45	18·32	19 18	20 03	20·89	21·75	22·61	23·47	24·32
40	14·53	14 69	15 38	16 24	17·10	17·97	18·83	19 69	20·55	21 42	22·28	23·14	24·00
45	14·16	14·33	15·02	15·89	16 75	17·62	18·49	19·35	20·21	21 08	21·95	22·82	23·68
50	13 79	13 96	14·66	15·52	16·39	17·26	18·14	19·01	19·87	20 75	21·61	22·49	23·35
55	13·42	13 59	14 29	15·16	16 03	16·91	17·78	18·66	19·52	20·40	21 27	22·15	23·02
60	13·05	13 22	13 92	14·79	15 67	16·55	17·43	18·31	19 18	20·06	20·93	21 82	22·69
62	12·90	13 07	13·77	14·65	15·52	16·41	17·29	18·17	19·04	19·92	20 80	21·68	22·56
65	12·67	12 85	13·55	14 43	15 30	16·19	17·07	17·95	18·83	19 71	20·59	21·48	22 35
70	12 29	12·47	13·17	14·05	14·93	15·82	16·71	17·59	18·47	19·36	20·25	21 14	22·02
75	11 91	12·08	12·79	13·68	14·56	15·45	16·35	17·23	18 12	19 01	19·90	20·79	21·67
80	11 52	11 69	12·41	13·30	14 18	15·08	15·98	16 87	17 76	18·65	19·54	20·44	21·33
85	11·13	11·30	12·02	12 91	13·81	14·71	15·61	16·50	17 39	18·30	19·19	20·09	20·98
90	10 72	10·90	11·62	12·52	13·42	14·32	15 22	16·12	17 02	17·92	18·82	19·73	20·62
95	10·32	10·51	11·23	12·13	13·03	13·94	14·85	15 75	16·65	17·56	18 46	19 37	20·27
100	9·93	10·10	10·83	11·74	12·64	13·55	14·47	15·37	16 28	17 19	18·09	19 01	19 91
110	9 11	9·29	10·02	10·99	11·85	12·77	13·69	14 61	15·52	16·44	17·36	18·28	19·19
120	8·23	8·46	9·20	10 12	11·04	11·97	12·91	13 83	14·75	15 68	16·60	17·53	18·45
130	7·43	7·61	8·36	9 29	10 22	11·16	12 10	13·03	13·96	14·90	15·82	16·76	17·69
140	6·57	6·75	7·51	8 44	9·38	10·33	11·28	12·22	13 16	14 10	15·04	15·99	16·93
150	5·63	5·87	6·63	7·58	8·52	9·48	10·44	11·39	12·33	13 29	14·24	15·19	16·14
160	4·78	4 97	5·74	6·69	7 65	8·62	9·58	10·54	11 50	12 46	13 42	14·38	15·34
170	3·87	4·06	4·84	5·80	6·77	7·74	8·72	9 68	10·65	11·62	12·59	13·57	14·53
180	2·93	3·13	3·91	4·88	5·86	6·84	7·83	8 80	9·78	10·76	11 74	12·72	13·70
190	1·98	2·17	2·95	3·95	4·93	5·93	6·92	7·90	8 89	9·88	10·87	11·86	12·85
200	1·00	1·20	2·00	2·99	3·99	4·99	6·00	6·99	7·98	8·99	9·98	10·99	11·93
210	·00	·21	1·00	2·01	3 01	4·03	5·04	6·05	7·05	8·06	9·07	10·09	11·09
212	—·21	·00	·81	1·81	2·82	3·84	4·85	5·86	6·87	7·88	8·89	9·91	10·91
220	1·00	—·81	·00	1·01	2·03	3·05	4·08	5·09	6·11	7·13	8·14	9·17	10·19
230	2·01	1·81	-1·01	·00	1·03	2·06	3·10	4·12	5·15	6·18	7·21	8·24	9·27
240	3·01	2·82	2·03	-1·03	·00	1·05	2·09	3·13	4·16	5·21	6·25	7·29	8·33
250	4·03	3 84	3·05	2·06	-1·05	·00	1·06	2·10	3·15	4·21	5 25	6·31	7·36
260	5·04	4·85	4·08	3·10	2·09	-1·06	·00	1·06	2·12	3·18	4·24	5·31	6·37
270	6·05	5 86	5·09	4 12	3·13	2·10	-1·06	·00	1·07	2·15	3·22	4·30	5·37
280	7·05	6·87	6 11	5·15	4·16	3·15	2·12	-1·07	·00	1·09	2·17	3·25	4·34
290	8·06	7·88	7·13	6·18	5·21	4·21	3·18	2·15	-1·09	·00	1·09	2·20	3·29
300	9·07	8·89	8·14	7 21	6·25	5·25	4·24	3·22	2·17	-1·09	·00	1 12	2·22
310	10·09	9 91	9·17	8·24	7·29	6·31	5·31	4·30	3·25	2·20	-1·12	·00	1·12
320	11·09	10·91	10.19	9 27	8 33	7 36	6·37	5·37	4·34	3·29	2·22	-1·12	·00

Table No. 7.

Areas and Circumferences of Circles from 1/64 to 4 inches in diameter varying by sixteenths and from 4 inches to 100 inches varying by one-eighth inch.

Diam. in Inches.	Area in Square Inches.	Circum. in Inches.	Diam. in Inches.	Area in Square Inches.	Circum. in Inches.	Diam. in Inches.	Area in Square Inches.	Circum. in Inches.
.$\frac{1}{64}$	0.00019	0.0490	3.$\frac{1}{16}$	7.3662	9.6211	8.$\frac{3}{8}$	55.088	26.31
.$\frac{1}{32}$	0.00076	0.0951	.$\frac{1}{8}$	7.6699	9.8175	.$\frac{1}{2}$	56.745	26.70
.$\frac{1}{16}$	0.00306	0.1963	.$\frac{3}{16}$	7.9798	10.0138	.$\frac{5}{8}$	58.426	27.10
.$\frac{1}{8}$	0.0122	0.3927	.$\frac{1}{4}$	8.2957	10.2102	.$\frac{3}{4}$	60.132	27.49
.$\frac{3}{16}$	0.0276	0.5890	.$\frac{5}{16}$	8.6179	10.4065	.$\frac{7}{8}$	61.862	27.88
.$\frac{1}{4}$	0.0490	0.7854	.$\frac{3}{8}$	8.9462	10.6029	9.	63.617	28.27
.$\frac{5}{16}$	0.0767	0.9817	.$\frac{7}{16}$	9.2806	10.7992	.$\frac{1}{8}$	65.396	28.66
.$\frac{3}{8}$	0.1104	1.1781	$\frac{1}{2}$	9.6211	10.9956	.$\frac{1}{4}$	67 200	29.06
.$\frac{7}{16}$	0.1503	1.3744	.$\frac{9}{16}$	9.9678	11.1919	.$\frac{3}{8}$	69.029	29.45
.$\frac{1}{2}$	0.1963	1.5708	.$\frac{5}{8}$	10.3210	11.3883	.$\frac{1}{2}$	70.882	29.85
.$\frac{9}{16}$	0.2485	1.7671	.$\frac{11}{16}$	10.6796	11.5846	.$\frac{5}{8}$	72.759	30.24
.$\frac{5}{8}$	0.3067	1.9630	.$\frac{3}{4}$	10.9446	11.7810	.$\frac{3}{4}$	74.662	30.63
.$\frac{11}{16}$	0.3712	2.1590	.$\frac{13}{16}$	11.4159	11.9773	.$\frac{7}{8}$	76.588	31.02
.$\frac{3}{4}$	0.4417	2.3565	.$\frac{7}{8}$	11.7932	12.1737	10.	78.540	31.42
.$\frac{13}{16}$	0.5174	2.5512	.$\frac{15}{16}$	12.1768	12.3700	.$\frac{1}{8}$	80.515	31.81
.$\frac{7}{8}$	0.6013	2.7490	4.	12.566	12.57	.$\frac{1}{4}$	82.516	32.20
.$\frac{15}{16}$	0.6902	2.9453	.$\frac{1}{8}$	13.364	12.96	.$\frac{3}{8}$	84.540	32.59
1.	0.7854	3.1416	.$\frac{1}{4}$	14.186	13.35	.$\frac{1}{2}$	86.590	32.99
.$\frac{1}{16}$	0.8861	3.3379	.$\frac{3}{8}$	15.033	13.74	.$\frac{5}{8}$	88.664	33.38
.$\frac{1}{8}$	0.9940	3.5343	.$\frac{1}{2}$	15.904	14.14	.$\frac{3}{4}$	90.762	33.77
.$\frac{3}{16}$	1.1075	3.7306	.$\frac{5}{8}$	16.800	14.53	.$\frac{7}{8}$	92.885	34.16
.$\frac{1}{4}$	1 2271	3.9270	.$\frac{3}{4}$	17.720	14.92	11.	95.033	34.56
.$\frac{5}{16}$	1.3529	4.1233	.$\frac{7}{8}$	18.665	15.32	.$\frac{1}{8}$	97 205	34.95
.$\frac{3}{8}$	1.4848	4.3197	5.	19.635	15.71	.$\frac{1}{4}$	99.402	35.34
.$\frac{7}{16}$	1.6229	4.5160	.$\frac{1}{8}$	20.629	16.10	.$\frac{3}{8}$	101.62	35.74
.$\frac{1}{2}$	1.7671	4.7124	.$\frac{1}{4}$	21.648	16.49	.$\frac{1}{2}$	103.87	36.13
.$\frac{9}{16}$	1.9175	4.9087	.$\frac{3}{8}$	22.690	16.89	.$\frac{5}{8}$	106.14	36.52
.$\frac{5}{8}$	2.0739	5.1051	.$\frac{1}{2}$	23.758	17.28	.$\frac{3}{4}$	108.43	36.91
.$\frac{11}{16}$	2.2365	5.3014	.$\frac{5}{8}$	24.850	17.67	.$\frac{7}{8}$	110.75	37.31
.$\frac{3}{4}$	2.4052	5.4978	.$\frac{3}{4}$	25.967	18.06	12.	113.10	37.70
.$\frac{13}{16}$	2.5801	5.6941	.$\frac{7}{8}$	27.108	18.46	.$\frac{1}{8}$	115.47	38.09
.$\frac{7}{8}$	2.7611	5.8905	6.	28.274	18.85	.$\frac{1}{4}$	117.86	38.48
.$\frac{15}{16}$	2.9483	6.0868	.$\frac{1}{8}$	29.464	19.24	.$\frac{3}{8}$	120.28	38.88
2.	3.1416	6.2832	.$\frac{1}{4}$	30.680	19.64	.$\frac{1}{2}$	122.72	39.27
.$\frac{1}{16}$	3.3411	6.4795	.$\frac{3}{8}$	31.919	20.03	.$\frac{5}{8}$	125.18	39.66
.$\frac{1}{8}$	3.5468	6.6759	.$\frac{1}{2}$	33.183	20.42	.$\frac{3}{4}$	127.68	40.06
.$\frac{3}{16}$	3.7582	6.8722	.$\frac{5}{8}$	34.471	20.81	.$\frac{7}{8}$	130.19	40.45
.$\frac{1}{4}$	3.9760	7.0686	.$\frac{3}{4}$	35.785	21.21	13.	132.73	40.84
.$\frac{5}{16}$	4.2001	7.2649	.$\frac{7}{8}$	37.122	21.60	.$\frac{1}{8}$	135.30	41.23
.$\frac{3}{8}$	4.4302	7.4618	7.	38.484	21.99	.$\frac{1}{4}$	137.89	41.63
.$\frac{7}{16}$	4.6664	7.6576	.$\frac{1}{8}$	39.871	22.38	.$\frac{3}{8}$	140.50	42.02
.$\frac{1}{2}$	4.9087	7.8540	.$\frac{1}{4}$	41.282	22.78	.$\frac{1}{2}$	143.14	42.41
.$\frac{9}{16}$	5.1573	8.0503	.$\frac{3}{8}$	42.718	23.17	.$\frac{5}{8}$	145 80	42.80
.$\frac{5}{8}$	5.4119	8.2467	.$\frac{1}{2}$	44.179	23.56	.$\frac{3}{4}$	148.49	43.20
.$\frac{11}{16}$	5.6727	8.4430	.$\frac{5}{8}$	45.663	23.95	.$\frac{7}{8}$	151.20	43.59
.$\frac{3}{4}$	5.9395	8.6394	.$\frac{3}{4}$	47.173	24.35	14.	153.94	43.98
.$\frac{13}{16}$	6.2126	8.8357	.$\frac{7}{8}$	48.707	24.74	.$\frac{1}{8}$	156.70	44.38
.$\frac{7}{8}$	6.4918	9.0321	8.	50.265	25.13	.$\frac{1}{4}$	159.48	44.77
.$\frac{15}{16}$	6.7772	9.2284	.$\frac{1}{8}$	51.848	25.52	.$\frac{3}{8}$	162.29	45.16
3.	7.0686	9.4248	.$\frac{1}{4}$	53.456	25.92	.$\frac{1}{2}$	165.13	45.55

Table No. 7.—Continued.

Areas and Circumferences of Circles from 1-64 to 4 inches in diameter, varying by sixteenths; and from 4 inches to 100 inches varying by one-eighth inch.

Diam. in Inches.	Area in Square Inches.	Circum. in Inches.	Diam. in Inches.	Area in Square Inches.	Circum. in Inches.	Diam. in Inches.	Area in Square Inches.	Circum. in Inches.
14.5/8	167.99	45.95	21.3/8	358.84	67.15	28.1/8	621.26	88.36
.3/4	170.87	46.34	.1/2	363.05	67.54	.1/4	626.80	88.75
.7/8	173.78	46.73	.5/8	367.28	67.94	.3/8	632.36	89.14
15.	176.71	47.12	.3/4	371.54	68.33	.1/2	637.94	89.54
.1/8	179.67	47.52	.7/8	375.83	68.72	.5/8	643.55	89.93
.1/4	182.65	47.91	22.	380.13	69.12	.3/4	649.18	90.32
.3/8	185.66	48.30	.1/8	384.46	69.51	.7/8	654.84	90.71
.1/2	188.69	48.69	.1/4	388.82	69.90	29.	660.52	91.11
.5/8	191.75	49.09	.3/8	393.20	70.29	.1/8	666.23	91.50
.3/4	194.83	49.48	.1/2	397.61	70.69	.1/4	671.96	91.89
.7/8	197.93	49.87	.5/8	402.04	71.08	.3/8	677.71	92.28
16.	201.06	50.27	.3/4	406.49	71.47	.1/2	683.49	92.68
.1/8	204.22	50.66	.7/8	410.97	71.86	.5/8	689.30	93.07
.1/4	207.39	51.05	23.	415.48	72.26	.3/4	695.13	93.46
.3/8	210.60	51.44	.1/8	420.	72.65	.7/8	700.98	93.85
.1/2	213.82	51.84	.1/4	424.56	73.04	30.	706.86	94.25
.5/8	217.08	52.23	.3/8	429.13	73.43	.1/8	712.76	94.64
.3/4	220.35	52.62	.1/2	433.74	73.83	.1/4	718.69	95.03
.7/8	223.65	53.01	.5/8	438.36	74.22	.3/8	724.64	95.43
17.	226.98	53.41	.3/4	443.01	74.61	.1/2	730.62	95.82
.1/8	230.33	53.80	.7/8	447.70	75.	.5/8	736.62	96.21
.1/4	233.70	54.19	24.	452.39	75.40	.3/4	742.64	96.60
.3/8	237.10	54.59	.1/8	457.11	75.79	.7/8	748.69	97.
.1/2	240.53	54.98	.1/4	461.86	76.18	31.	754.77	97.39
.5/8	243.98	55.37	.3/8	466.64	76.58	.1/8	760.87	97.78
.3/4	247.45	55.76	.1/2	471.44	76.97	.1/4	766.99	98.17
.7/8	250.95	56.16	.5/8	476.26	77.36	.3/8	773.14	98.57
18.	254.47	56.55	.3/4	481.11	77.75	.1/2	779.31	98.97
.1/8	258.02	56.94	.7/8	485.98	78.15	.5/8	785.51	99.35
.1/4	261.59	57.33	25.	490.87	78.54	.3/4	791.73	99.75
.3/8	265.18	57.73	.1/8	495.80	78.93	.7/8	797.98	100.14
.1/2	268.80	58.12	.1/4	500.74	79.33	32.	804.25	100.53
.5/8	272.45	58.51	.3/8	505.71	79.72	.1/8	810.54	100.92
.3/4	276.12	58.90	.1/2	510.71	80.11	.1/4	816.86	101.32
.7/8	279.81	59.30	.5/8	515.72	80.50	.3/8	823.21	101.71
19.	283.53	59.69	.3/4	520.77	80.90	.1/2	829.58	102.10
.1/8	287.27	60.08	.7/8	525.84	81.29	.5/8	835.97	102.49
.1/4	291.04	60.48	26.	530.93	81.68	.3/4	842.39	102.89
.3/8	294.83	60.87	.1/8	536.05	82.07	.7/8	848.83	103.28
.1/2	298.65	61.26	.1/4	541.19	82.47	33.	855.30	103.67
.5/8	302.49	61.65	.3/8	546.36	82.86	.1/8	861.79	104.06
.3/4	306.35	62.05	.1/2	551.55	83.25	.1/4	868.30	104.46
.7/8	310.25	62.44	.5/8	556.76	83.64	.3/8	874.84	104.85
20.	314.16	62.83	.3/4	562.	84.04	.1/2	881.41	105.24
.1/8	318.10	63.22	.7/8	567.27	84.43	.5/8	888.	105.64
.1/4	322.06	63.62	27.	572.56	84.82	.3/4	894.62	106.03
.3/8	326.05	64.01	.1/8	577.87	85.21	.7/8	901.25	106.42
.1/2	330.06	64.40	.1/4	583.21	85.61	34.	907.92	106.81
.5/8	334.10	64.79	.3/8	588.57	86.	.1/8	914.61	107.21
.3/4	338.16	65.19	.1/2	593.96	86.39	.1/4	921.32	107.60
.7/8	342.25	65.58	.5/8	599.37	86.79	.3/8	928.06	107.99
21.	346.36	65.97	.3/4	604.81	87.18	.1/2	934.82	108.39
1/8	350.50	66.37	.7/8	610.27	87.57	.5/8	941.60	108.78
1/4	354.66	66.76	28.	615.75	87.96	.3/4	948.42	109.17

Table No. 7.—Continued.

Areas and Circumferences of Circles from 1-64 to 4 inches in diameter varying by sixteenths; and from 4 inches to 100 inches varying by one-eighth inch.

Diam. in Inches.	Area in Square Inches.	Circum. in Inches.	Diam. in Inches.	Area in Square Inches.	Circum. in Inches.	Diam. in Inches.	Area in Square Inches.	Circum. in Inches.
34.7/8	955.25	109.56	41.5/8	1360.8	130.8	48.3/8	1837.9	152.
35.	962.11	109.96	.3/4	1369.	131.2	.1/2	1847.5	152.4
.1/8	968.99	110.35	.7/8	1377.2	131.6	.5/8	1857.	152.8
.1/4	975.91	110.74	42.	1385.4	131.9	.3/4	1866.5	153.2
.3/8	982.84	111.13	.1/8	1393.7	132.3	.7/8	1876.1	153.5
.1/2	989.80	111.53	.1/4	1402.	132.7	49.	1885.7	153.9
.5/8	996.78	111.92	.3/8	1410.3	133.1	.1/8	1895.4	154.3
.3/4	1003.79	112.31	.1/2	1418.6	133.5	.1/4	1905.	154.7
.7/8	1010.80	112.70	.5/8	1427.	133.9	.3/8	1914.7	155.1
36.	1017.88	113.10	.3/4	1435.4	134.3	.1/2	1924.4	155.5
.1/8	1024.95	113.49	.7/8	1443.8	134.7	.5/8	1934.1	155.9
.1/4	1032.06	113.88	43.	1452.2	135.1	.3/4	1943.9	156.3
.3/8	1039.19	114.28	.1/8	1460.6	135.5	.7/8	1953.7	156.7
.1/2	1046.35	114.67	.1/4	1469.1	135.9	50.	1963.5	157.1
.5/8	1053.52	115.06	.3/8	1477.6	136.3	.1/8	1973.3	157.4
.3/4	1060.73	115.45	.1/2	1486.2	136.7	.1/4	1983.2	157.9
.7/8	1067.95	115.85	.5/8	1494.7	137.1	.3/8	1993.	158.2
37.	1075.2	116.2	.3/4	1503.3	137.4	.1/2	2003.	158.7
.1/8	1082.5	116.6	.7/8	1511.9	137.8	.5/8	2012.8	159.
.1/4	1089.8	117.	44.	1520.5	138.2	.3/4	2022.8	159.4
.3/8	1097.1	117.4	.1/8	1529.2	138.6	.7/8	2032.8	159.8
.1/2	1104.5	117.8	.1/4	1537.9	139.	51.	2042.8	160.2
.5/8	1111.8	118.2	.3/8	1546.5	139.4	.1/8	2052.8	160.6
.3/4	1119.2	118.6	.1/2	1555.3	139.8	.1/4	2062.9	161.
.7/8	1126.7	119.	.5/8	1564.	140.2	.3/8	2072.9	161.3
38.	1134.1	119.4	.3/4	1572.8	140.6	.1/2	2083.1	161.8
.1/8	1141.6	119.8	.7/8	1581.6	141.	.5/8	2093.2	162.1
.1/4	1149.1	120.2	45.	1590.4	141.4	.3/4	2103.3	162.6
.3/8	1156.6	120.6	.1/8	1599.3	141.8	.7/8	2113.5	162.9
.1/2	1164.2	121.	.1/4	1608.2	142.2	52.	2123.7	163.4
.5/8	1171.7	121.3	.3/8	1617.	142.6	.1/8	2133.9	163.7
.3/4	1179.3	121.7	.1/2	1626.	142.9	.1/4	2144.2	164.1
.7/8	1186.9	122.1	.5/8	1634.9	143.3	.3/8	2154.4	164.5
39.	1194.6	122.5	.3/4	1643.9	143.7	.1/2	2164.8	164.9
.1/8	1202.3	122.9	.7/8	1652.9	144.1	.5/8	2175.	165.3
.1/4	1210.	123.3	46.	1661.9	144.5	.3/4	2185.4	165.7
.3/8	1217.7	123.7	.1/8	1671.	144.9	.7/8	2195.7	166.1
.1/2	1225.4	124.1	.1/4	1680.	145.3	53.	2206.2	166.5
.5/8	1233.2	124.5	.3/8	1689.1	145.7	.1/8	2216.6	166.8
.3/4	1241.	124.9	.1/2	1698.2	146.1	.1/4	2227.	167.3
.7/8	1248.8	125.3	.5/8	1707.4	146.5	.3/8	2237.5	167.6
40.	1256.6	125.6	.3/4	1716.5	146.9	.1/2	2248.	168.1
.1/8	1264.5	126.	.7/8	1725.7	147.3	.5/8	2258.5	168.4
.1/4	1272.4	126.4	47.	1734.9	147.7	.3/4	2269.	168.9
.3/8	1280.3	126.8	.1/8	1744.2	148.	.7/8	2279.6	169.2
.1/2	1288.2	127.2	.1/4	1753.5	148.4	54.	2290.2	169 6
.5/8	1296.2	127.6	.3/8	1762.7	148.8	.1/8	2300.8	170.
.3/4	1304.2	128.	.1/2	1772.1	149.2	.1/4	2311.5	170.4
.7/8	1312.2	128.4	.5/8	1781.4	149.6	.3/8	2322.1	170.8
41.	1320.3	128.8	.3/4	1790.8	150.	.1/2	2332.8	171.2
.1/8	1328.3	129.2	.7/8	1800.1	150.4	.5/8	2343.5	171.6
.1/4	1336.4	129.6	48.	1809.6	150.8	.3/4	2354.3	172.
.3/8	1344.5	130.	.1/8	1819.	151.2	.7/8	2365.	172.3
.1/2	1352.7	130.4	.1/4	1828.5	151.6	55.	2375.8	172.8

TABLE NO. 7.—CONTINUED.

Areas and Circumferences of Circles from 1-64 to 4 inches in diameter varying by sixteenths; and from 4 inches to 100 inches, varying by one-eighth inch.

Diam. in Inches.	Area in Square Inches.	Circum. in Inches.	Diam in Inches.	Area in Square Inches.	Circum. in Inches.	Diam. in Inches.	Area in Square Inches.	Circum. in Inches.
55.1/8	2386.6	173.1	61.7/8	3006.9	194.3	68.5/8	3698.7	215.5
.1/4	2397.5	173.6	62.	3019.1	194.8	.3/4	3712.2	215.9
.3/8	2408.3	173.9	.1/8	3031.2	195.1	.7/8	3725.7	216.3
.1/2	2419.2	174.4	.1/4	3043.5	195.6	69.	3739.3	216.7
.5/8	2430.1	174.7	.3/8	3055.7	195.9	.1/8	3752.8	217.1
.3/4	2441.	175.1	.1/2	3068.	196.3	.1/4	3766.4	217.5
.7/8	2452.	175.5	.5/8	3080.2	196.7	.3/8	3780.	217.9
56.	2463.	175.9	.3/4	3092.6	197.1	.1/2	3793.7	218.3
.1/8	2474.	176.3	.7/8	3104.8	197.5	.5/8	3807.3	218.7
.1/4	2485.	176.7	63.	3117.2	197.9	.3/4	3821.	219.1
.3/8	2496.1	177.1	.1/8	3129.6	198.3	.7/8	3834.7	219.5
.1/2	2507.2	177.5	.1/4	3142.	198.7	70.	3848.5	219.9
.5/8	2518.2	177.8	.3/8	3154.4	199.	.1/8	3862.2	220.3
.3/4	2529.4	178.3	.1/2	3166.9	199.5	.1/4	3876.	220.7
.7/8	2540.5	178.6	.5/8	3179.4	199.8	.3/8	3889.8	221.
57.	2551.8	179.1	.3/4	3191.9	200.3	.1/2	3903.6	221.5
.1/8	2562.9	179.4	.7/8	3204.4	200.6	.5/8	3917.4	221.8
.1/4	2574.2	179.9	64.	3217.	201.1	.3/4	3931.4	222.2
.3/8	2585.4	180.2	.1/8	3229.5	201.4	.7/8	3945.2	222.6
.1/2	2596.7	180.6	.1/4	3242.2	201.8	71.	3959.2	223.
.5/8	2608.	181.	.3/8	3254.8	202.2	.1/8	3973.1	223.4
.3/4	2619.4	181.4	.1/2	3267.5	202.6	.1/4	3987.1	223.8
.7/8	2630.7	181.8	.5/8	3280.1	203.	.3/8	4001.1	224.2
58.	2642.1	182.2	.3/4	3292.8	203.4	.1/2	4015.2	224.6
.1/8	2653.4	182.6	.7/8	3305.5	203.8	.5/8	4029.2	225.
.1/4	2664.9	183.	65.	3318.3	204.2	.3/4	4043.3	225.4
.3/8	2676.3	183.3	.1/8	3331.	204.5	.7/8	4057.	225.8
.1/2	2687.8	183.8	.1/4	3343.9	205.	72.	4071.5	226.2
.5/8	2699.3	184.1	.3/8	3356.7	205.3	.1/8	4085.6	226.5
.3/4	2710.9	184.6	.1/2	3369.6	205.8	.1/4	4099.8	227.
.7/8	2722.4	184.9	.5/8	3382.4	206.1	.3/8	4114.	227.3
59.	2734.	185.4	.3/4	3395.3	206.6	.1/2	4128.2	227.7
.1/8	2745.5	185.7	.7/8	3408.2	206.9	.5/8	4142.5	228.1
.1/4	2757.2	186.1	66.	3421.2	207.3	.3/4	4156.8	228.5
.3/8	2768.8	186.5	.1/8	3434.1	207.7	.7/8	4171.	228.9
.1/2	2780.5	186.9	.1/4	3447.2	208.1	73.	4185.4	229.3
.5/8	2792.2	187.3	.3/8	3460.1	208.5	.1/8	4199.7	229.7
.3/4	2803.9	187.7	.1/2	3473.2	208.9	.1/4	4214.1	230.1
.7/8	2815.6	188.1	.5/8	3486.3	209.3	.3/8	4228.5	230.5
60.	2827.4	188.5	.3/4	3499.4	209.7	.1/2	4242.9	230.9
.1/8	2839.2	188.8	.7/8	3512.5	210.	.5/8	4257.3	231.3
.1/4	2851.	189.3	67.	3525.6	210.5	.3/4	4271.8	231.7
.3/8	2862.8	189.6	.1/8	3538.8	210.8	.7/8	4286.3	232.
.1/2	2874.8	190.1	.1/4	3552.	211.3	74.	4300.8	232.5
.5/8	2886.6	190.4	.3/8	3565.2	211.6	.1/8	4315.3	232.8
.3/4	2898.5	190.9	.1/2	3578.5	212.1	.1/4	4329.9	233.2
.7/8	2910.6	191.2	.5/8	3591.7	212.4	.3/8	4344.5	233.6
61.	2922.5	191.6	.3/4	3605.	212.8	.1/2	4359.2	234.
.1/8	2934.4	192.	.7/8	3618.3	213.2	.5/8	4373.8	234.4
.1/4	2946.5	192.4	68.	3631.7	213.6	.3/4	4388.5	234.8
.3/8	2958.5	192.8	.1/8	3645.	214.	.7/8	4403.1	235.2
.1/2	2970.6	193.2	.1/4	3658.4	214.4	75.	4417.9	235.6
.5/8	2982.6	193.6	.3/8	3671.8	214.8	.1/8	4432.6	236.
.3/4	2994.8	194.	.1/2	3685.3	215.2	.1/4	4447.4	236.4

TABLE NO. 7.—CONTINUED.

Areas and Circumferences of Circles from 1-64 to 4 inches in diameter varying by sixteenths; and from 4 inches to 100 inches, varying by one-eighth inch.

Diam. in Inches.	Area in Square Inches.	Circum. in Inches.	Diam. in Inches.	Area in Square Inches.	Circum. in Inches.	Diam. in Inches.	Area in Square Inches.	Circum. in Inches.
75.3/8	4462.1	236.7	82.1/8	5297.1	258.	88.7/8	6203.6	279.2
.1/2	4477.	237.2	.1/4	5313.3	258.4	89.	6221.1	279.6
.5/8	4491.8	237.5	.3/8	5329.4	258.8	.1/8	6238.6	280.
.3/4	4506.7	238.	.1/2	5345.6	259.2	.1/4	6256.1	280.4
.7/8	4521.5	238.3	.5/8	5361.8	259.6	.3/8	6273.6	280.8
76.	4536.5	238.8	.3/4	5378.1	260.	.1/2	6291.2	281.2
.1/8	4551.4	239.1	.7/8	5394.3	260.4	.5/8	6308.8	281.6
.1/4	4566.4	239.5	83.	5410.6	260.8	.3/4	6326.4	282.
.3/8	4581.3	239.9	.1/8	5426.9	261.1	.7/8	6344.	282.3
.1/2	4596.3	240.3	.1/4	5443.3	261.5	90.	6361.7	282.7
.5/8	4611.3	240.7	.3/8	5459.6	261.9	.1/8	6379.4	283.1
.3/4	4626.4	241.1	.1/2	5476.	262.3	.1/4	6397.1	283.5
.7/8	4641.5	241.5	.5/8	5492.4	262.7	.3/8	6414.8	283.9
77.	4656.6	241.9	.3/4	5508.8	263.1	.1/2	6432.6	284.3
.1/8	4671.7	242.2	.7/8	5525.3	263.5	.5/8	6450.4	284.7
.1/4	4686.9	242.7	84.	5541.8	263.9	.3/4	6468.2	285.1
.3/8	4702.1	243.	.1/8	5558.3	264.3	.7/8	6486.	285.5
.1/2	4717.3	243.5	.1/4	5574.8	264.7	91.	6503.9	285.9
.5/8	4732.5	243.8	.3/8	5591.3	265.	.1/8	6521.7	286.3
.3/4	4747.8	244.3	.1/2	5607.9	265.5	.1/4	6539.7	286.7
.7/8	4763.	244.6	.5/8	5624.5	265.8	.3/8	6557.6	287.1
78.	4778.4	245.	.3/4	5641.2	266.2	.1/2	6575.5	287.5
.1/8	4793.7	245.4	.7/8	5657.8	266.6	.5/8	6593.5	287.8
.1/4	4809.	245.8	85.	5674.5	267.	.3/4	6611.5	288.2
.3/8	4824.4	246.2	.1/8	5691.2	267.4	.7/8	6629.5	288.6
.1/2	4839.8	246.6	.1/4	5707.9	267.8	92.	6647.6	289.
.5/8	4855.2	247.	.3/8	5724.6	268.2	.1/8	6665.7	289.4
.3/4	4870.8	247.4	.1/2	5741.5	268.6	.1/4	6683.8	289.8
.7/8	4886.1	247.7	.5/8	5758.2	268.9	.3/8	6701.9	290.2
79.	4901.7	248.2	.3/4	5775.1	269.4	.1/2	6720.1	290.6
.1/8	4917.2	248.5	.7/8	5791.9	269.7	.5/8	6738.2	291.
.1/4	4932.7	249.	86.	5808.8	270.2	.3/4	6756.4	291.4
.3/8	4948.3	249.3	.1/8	5825.7	270.5	.7/8	6774.7	291.8
.1/2	4963.9	249.8	.1/4	5842.6	271.	93.	6792.9	292.2
.5/8	4979.5	250.1	.3/8	5859.5	271.3	.1/8	6811.1	292.6
.3/4	4995.2	250.5	.1/2	5876.5	271.7	.1/4	6829.5	293.
.7/8	5010.8	250.9	.5/8	5893.5	272.1	.3/8	6847.8	293.4
80.	5026.5	251.3	.3/4	5910.6	272.5	.1/2	6866.1	293.7
.1/8	5042.2	251.7	.7/8	5927.6	272.9	.5/8	6884.5	294.1
.1/4	5058.	252.1	87.	5944.7	273.3	.3/4	6902.9	294.5
.3/8	5073.7	252.5	.1/8	5961.7	273.7	.7/8	6921.3	294.9
.1/2	5089.6	252.9	.1/4	5978.9	274.1	94.	6939.8	295.3
.5/8	5105.4	253.3	.3/8	5996.	274.4	.1/8	6958.2	295.7
.3/4	5121.2	253.7	.1/2	6013.2	274.9	.1/4	6976.7	296.1
.7/8	5137.1	254.1	.5/8	6030.4	275.2	.3/8	6995.2	296.5
81.	5153.	254.5	.3/4	6047.6	275.7	.1/2	7013.8	296.9
.1/8	5168.9	254.9	.7/8	6064.8	276.	.5/8	7032.3	297.3
.1/4	5184.9	255.3	88.	6082.1	276.5	.3/4	7051.	297.7
.3/8	5200.8	255.6	.1/8	6099.4	276.8	.7/8	7069.5	298.1
.1/2	5216.8	256.	.1/4	6116.7	277.2	95.	7088.2	298.5
.5/8	5232.8	256.4	.3/8	6134.	277.6	.1/8	7106.9	298.8
.3/4	5248.9	256.8	.1/2	6151.4	278.	.1/4	7125.6	299.2
.7/8	5264.9	257.2	.5/8	6168.8	278.4	.3/8	7144.3	299.6
82.	5281.	257.6	.3/4	6186.2	278.8	.1/2	7163.	300.

TABLE NO. 7.—CONTINUED.

Areas and Circumferences of Circles from 1-64 to 4 inches in diameter, varying by sixteenths; and from 4 inches to 100 inches, varying by one-eighth inch.

Diam. in Inches.	Area in Square Inches.	Circum. in Inches.	Diam. in Inches.	Area in Square Inches.	Circum. in Inches.	Diam. in Inches.	Area in Square Inches.	Circum. in Inches.
95.$\frac{5}{8}$	7181.8	300.4	97.$\frac{1}{8}$	7408.8	305.1	98.$\frac{5}{8}$	7639.4	309.8
.$\frac{3}{4}$	7200.6	300.8	.$\frac{1}{4}$	7428.	305.5	.$\frac{3}{4}$	7658.9	310.2
.$\frac{7}{8}$	7219.4	301.2	.$\frac{3}{8}$	7447.	305.9	.$\frac{7}{8}$	7678.2	310.6
96.	7238.2	301.6	.$\frac{1}{2}$	7466.2	306.3	99.	7697.7	311.
.$\frac{1}{8}$	7257.1	302.	.$\frac{5}{8}$	7485.3	306.7	.$\frac{1}{8}$	7717.1	311.4
.$\frac{1}{4}$	7276.	302.4	.$\frac{3}{4}$	7504.5	307.1	.$\frac{1}{4}$	7736.6	311.8
.$\frac{3}{8}$	7294.9	302.8	.$\frac{7}{8}$	7523.7	307.5	.$\frac{3}{8}$	7756.1	312.2
.$\frac{1}{2}$	7313.8	303.2	98.	7543.	307.9	.$\frac{1}{2}$	7775.6	312.6
.$\frac{5}{8}$	7332.8	303.5	.$\frac{1}{8}$	7562.2	308.3	.$\frac{5}{8}$	7795.2	313.
.$\frac{3}{4}$	7351.8	303.9	.$\frac{1}{4}$	7581.5	308.7	.$\frac{3}{4}$	7814.8	313.4
.$\frac{7}{8}$	7370.7	304.3	.$\frac{3}{8}$	7600.8	309.	.$\frac{7}{8}$	7834.3	313.8
97.	7389.8	304.7	.$\frac{1}{2}$	7620.1	309.4	100.	7854.	314.2

If the areas of larger circles are required, they will be found by the following:

RULE—Multiply the square of the diameter in inches, by the decimal 0.7854, and the product will be the area in square inches; or, multiply half the circumference by half the diameter. If the circumference of a larger circle is wanted, and having the diameter, the rule is as follows:

RULE—As 7 is to 22, so is the diameter to the circumference, or diameter multiplied by 3.1416 equal circumference.

TABLE No. 8.

The properties of Saturated Steam.

Pressure per Square Inch.		Temperature in Fahrenheit Degrees.	Number of British Thermal Units in One Pound from Zero (Fahrenheit).			Weight of One Cubic Foot of Steam in Decimals of a Pound.	Number of Cubic Feet of Steam from One Cubic Foot of Water.
Total Pressure in Pounds from a Vacuum	Pressure in Pounds as Shown by Steam Gauge.		Number of Units of Heat in Water.	Number of Units of Heat Required for Evaporation, Called Latent Heat.	Total Number of Units of Heat Contained in Steam.		
1	....	102.	102.086	1042.964	1145.050	.0030	20620.0
2	....	126.266	126.440	1026.010	1152.450	.0058	10720.0
3	....	141.622	141.877	1015.254	1157.131	.0085	7326.0
4	......	153.070	153.396	1007.229	1160.625	.0112	5600.0
5	....	162.330	162.722	1000.727	1163.449	.0137	4535.0
6	....	170.123	170.577	995.249	1165.826	.0163	3814.0
7	....	176.910	177.425	990.471	1167.896	.0189	3300.0
8	....	182.910	183.481	986.245	1169.726	.0214	2910.0
9	....	188.316	188.941	982.434	1171.375	.0239	2607.0
10	....	193.240	193.919	978.958	1172.877	.0264	2360.0
11	....	197.768	198.496	975.762	1174.258	.0289	2157.0
12	....	201.960	202.737	972.800	1175.537	.0313	1988.0
13	....	205.885	206.709	970.025	1176.734	.0337	1846.0
14	....	209.560	210.428	967.427	1177.855	.0362	1722.0
14.7	....	212.000	212.900	965.700	1178.600	.03797	1644.0
15	.304	213.025	213.939	964.973	1178.912	.0387	1612.0
16	1.304	216.296	217.252	962.657	1179.909	.0413	1514.0
17	2.304	219.410	220.409	960.450	1180.859	.0437	1427.0
18	3.304	222.378	223.419	958.345	1181.764	.0462	1350.6
19	4.304	225.203	226.285	956.343	1182.628	.0487	1282.1
20	5.304	227.917	229.039	954.415	1183.454	.0511	1220.3
21	6.304	230.515	231.676	952.570	1184.246	.0536	1164.4
22	7.304	233.017	234.218	950.791	1185.009	.0561	1113.5
23	8.304	235.432	236.672	949.072	1185.744	.0585	1066.9
24	9.304	237.752	239.029	947.424	1186.453	.0610	1024.1
25	10.304	240.000	241.314	945.825	1187.139	.0634	984.8
26	11.304	242.175	243.526	944.277	1187.803	.0658	948.4
27	12.304	244.284	245.671	942.775	1188.446	.0683	914.6
28	13.304	246.326	247.748	941.321	1189.069	.0707	883.2
29	14.304	248.310	249.769	939.905	1189.674	.0731	854.0
30	15.304	250.245	251.738	938.925	1190.263	.0755	826.8
31	16.304	252.122	253.648	937.1878	1190.8358	.0779	801.2
32	17.304	253.952	255.512	935.8818	1191.3938	.0803	777.2
33	18.304	255.735	257.329	934.6088	1191.9378	.0827	754.7
34	19.304	257.476	259.103	933.3658	1192.4688	.0851	733.5
35	20.304	259.176	260.835	932.1523	1192.9873	.0875	713.4
36	21.304	260.835	262.527	930.9668	1193.4938	.0899	694.5
37	22.304	262.458	264.182	929.8068	1193.9888	.0922	676.6
38	23.304	264.045	265.801	928.6718	1194.4728	.0946	659.7
39	24.304	265.599	267.386	927.5608	1194.9468	.0970	643.6
40	25.304	267.120	268.938	926.4728	1195.4108	.0994	628.2
41	26.304	268.611	270.460	925.4058	1195.8658	.1017	613.4
42	27.304	270.073	271.954	924.3578	1196.3118	.1041	599.3
43	28.304	271.507	273.417	923.3323	1196.7493	.1064	586.1
44	29.304	272.915	274.855	922.3238	1197.1788	.1088	573.7
45	30.304	274.296	276.266	921.3343	1197.6003	.1111	561.8
46	31.304	275.652	277.651	920.3632	1198.0142	.1134	550.4
47	32.304	276.986	279.016	919.4052	1198.4212	.1158	539.5
48	33.304	278.297	280.355	918.4662	1198.8212	.1181	529.0
49	34.304	279.585	281.672	917.5422	1199.2142	.1204	518.6
50	35.304	280.854	282.969	916.6316	1199.6006	.1227	508.5
51	36.304	282.099	284.243	915.7377	1199.9807	.1251	499.1
52	37.304	283.326	285.499	914.8557	1200.3547	.1274	490.1

Table No. 8.—Continued.

The Properties of Saturated Steam.

Pressure per Square Inch.		Temperature in Fahrenheit Degrees.	Number of British Thermal Units in One Pound from Zero (Fahrenheit).			Weight of One Cubic Foot of Steam in Decimals of a Pound.	Number of Cubic Feet of Steam from One Cubic Foot of Water.
Total Pressure in Pounds from a Vacuum	Pressure in Pounds as Shown by Steam Gauge.		Number of Units of Heat in Water.	Number of Units of Heat Required for Evaporation, Called Latent Heat.	Total Number of Units of Heat Contained in Steam.		
53	38.304	284.534	286.736	913.9871	1200.7231	.1297	481.4
54	39.304	285.724	287.952	913.1340	1201.0860	.1320	472.9
55	40.304	286.897	289.153	912.2906	1201.4436	.1343	464.7
56	41.304	288.052	290.335	911.4611	1201.7961	.1366	457.0
57	42.304	289.112	291.503	910.6407	1202.1437	.1388	449.6
58	43.304	290.316	292.654	909.8325	1202.4865	.1411	442.4
59	44.304	291.425	293.790	909.0346	1202.8246	.1434	435.3
60	45.304	292.520	294.911	908.2472	1203.1582	.1457	428.5
61	46.304	293.598	296.016	907.4713	1203.4873	.14793	422.0
62	47.304	294.663	297.108	906.7042	1203.8122	.15021	415.6
63	48.304	295.714	298.185	905.9477	1204.1329	.15248	409.4
64	49.304	296.752	299.249	905.2005	1204.4495	.15471	403.5
65	50.304	297.777	300.300	904.4621	1204.7621	.15697	397.7
66	51.304	298.789	301.338	903.7327	1205.0707	.15921	392.1
67	52.304	299.789	302.364	903.0116	1205.3756	.16147	386.6
68	53.304	300.776	303.377	902.2999	1205.6769	.16372	381.3
69	54.304	301.753	304.380	901.5947	1205.9747	.16598	376.1
70	55.304	302.718	305.370	900.8991	1206.2691	.16817	371.2
71	56.304	303.673	306.350	900.2101	1206.5601	.17038	366.4
72	57.304	304.617	307.320	899.5280	1206.8480	.17259	361.7
73	58.304	305.551	308.279	898.8537	1207.1327	.17481	357.1
74	59.304	306.474	309.228	898.1863	1207.4143	.17704	352.6
75	60.304	307.388	310.166	897.5269	1207.6929	.17923	348.3
76	61.304	308.290	311.092	896.8764	1207.9684	.18142	344.1
77	62.304	309.184	312.011	896.2301	1208.2411	.18360	340.0
78	63.304	310.069	312.920	895.5910	1208.5110	.18579	336.0
79	64.304	310.945	313.821	894.9571	1208.7781	.18797	332.1
80	65.304	311.812	314.712	894.3304	1209.0424	.19015	328.3
81	66.304	312.670	315.595	893.7092	1209.3042	.19232	324.6
82	67.304	313.520	316.468	893.0954	1209.5634	.19454	320.9
83	68.304	314.361	317.333	892.4871	1209.8201	.19674	317.3
84	69.304	315.195	318.190	891.8843	1210.0743	.19887	313.9
85	70.304	316.021	319.040	891.2862	1210.3262	.20105	310.5
86	71.304	316.839	319.882	890.6938	1210.5758	.20321	307.2
87	72.304	317.650	320.717	890.1061	1210.8231	.20535	304.0
88	73.304	318.453	321.543	889.5251	1211.0681	.20753	300.8
89	74.304	319.249	322.362	888.9490	1211.3110	.20970	297.7
90	75.304	320.039	323.176	888.3758	1211.5518	.21183	294.7
91	76.304	320.821	323.981	887.8094	1211.7904	.21393	291.8
92	77.304	321.597	324.781	887.2460	1212.0270	.21608	288.9
93	78.304	322.366	325.572	886.6896	1212.2616	.21829	286.1
94	79.304	323.128	326.358	886.1362	1212.4942	.22045	283.3
95	80.304	323.884	327.136	885.5887	1212.7247	.22247	280.6
96	81.304	324.634	327.909	885.0444	1212.9534	.22455	278.0
97	82.304	325.378	328.675	884.5052	1213.1802	.22667	275.4
98	83.304	326.114	329.433	883.9721	1213.4051	.22883	272.8
99	84.304	326.845	330.186	883.4421	1213.6281	.23095	270.3
100	85.304	327.571	330.935	882.9144	1213.8494	.23302	267.9
101	86.304	328.291	331.678	882.3909	1214.0689	.23510	265.5
102	87.304	329.005	332.414	881.8727	1214.2867	.23717	263.2
103	88.304	329.714	333.145	881.3577	1214.5027	.23925	260.9
104	89.304	330.416	333.869	880.8481	1214.7171	.24132	258.7
105	90.304	331.113	334.587	880.3429	1214.9299	.24340	256.5

TABLE No. 8.—CONTINUED.

The Properties of Saturated Steam.

Pressure per Square Inch.		Temp-erature in Fahrenheit Degrees.	Number of British Thermal Units in One Pound from Zero (Fahrenheit).			Weight of One Cubic Foot of Steam in Decimals of a Pound.	Number of Cubic Feet of Steam from One Cubic Foot of Water.
Total Pressure in Pounds from a Vacuum	Pressure in Pounds as Shown by Steam Gauge.		Number of Units of Heat in Water.	Number of Units of Heat Required for Evaporation, Called Latent Heat.	Total Number of Units of Heat Contained in Steam.		
106	91.304	331.805	335.301	879.8400	1215.1410	.24547	254.3
107	92.304	332.492	336.009	879.3416	1215.3506	.24754	252.2
108	93.304	333.174	336.714	878.8447	1215.5587	.24961	250.1
109	94.304	333.851	337.411	878.3542	1215.7652	.25168	248.0
110	95.304	334.523	338.105	877.8653	1215.9703	.25376	246.0
111	96.304	335.191	338.795	877.3789	1216.1739	.25582	244.0
112	97.304	335.854	339.479	876.8970	1216.3760	.25788	242.0
113	98.304	336.511	340.157	876.4198	1216.5768	.25994	240.1
114	99.304	337 165	340.832	875.9442	1216.7762	.26199	238.2
115	100.304	337.814	341.502	875.4721	1216.9741	.26405	236.3
116	101.304	338.459	342.169	875.0018	1217.1708	.26611	234.5
117	102.304	339.100	342.831	874.5352	1217.3662	.26816	232.7
118	103.304	339.736	343.488	874.0722	1217.5602	.27020	231.0
119	104.304	340.368	344.141	873.6120	1217.7530	.27224	229.3
120	105.304	340.995	344.789	873.1555	1217.9445	.27428	227.6
121	106.304	341.618	345.432	872.7027	1218.1347	.27628	226.0
122	107.304	342.238	346.073	872.2508	1218.3238	.27828	224.4
123	108.304	342.854	346.709	871.8027	1218.5117	.28027	222.8
124	109.304	343.466	347.343	871.3553	1218.6983	.28227	221.2
125	110.304	344.074	347.972	870.9118	1218.8838	.28426	219.7
126	111.304	344.678	348.596	870.4721	1219.0681	.28625	218.2
127	112.304	345.279	349.217	870.0342	1219.2512	.28824	216.7
128	113.304	345.876	349.833	869.5983	1219.4333	.29023	215.2
129	114.304	346.459	350.448	869.1663	1219.6143	.29222	213.7
130	115.304	347.059	351.059	868.7351	1219.7941	.29420	212.3
131	116.304	347.644	351.665	868.3079	1219.9729	.29618	210.9
132	117.304	348.227	352.267	867.8836	1220.1506	.29816	209.5
133	118.304	348.806	352.867	867.4601	1220.3271	.30013	208.1
134	119.304	349.382	353.463	867.0397	1220.5027	.30209	206.7
135	120.304	349.954	354.055	866.6223	1220.6773	.30405	205.4
136	121.304	350.523	354.644	866.2068	1220.8508	.30601	204.1
137	122.304	351.089	355.230	866.7934	1221.0234	.30796	202.8
138	123.304	351.752	355.813	865.3820	1221.1950	.30990	201.5
139	124.304	352.211	356.392	864.9735	1221.3655	.31186	200.2
140	125.304	352.767	356.969	864.5661	1221.5351	.31386	199.0
141	126.304	353.319	357.541	864.1627	1221.7037	.31587	197.8
142	127.304	353.869	358.110	863.7613	1221.8713	.31788	196.6
143	128.304	354.416	358.677	863.3611	1222.0381	.31990	195.4
144	129.304	354.960	359.240	862.9640	1222.2040	.32190	194.2
145	130.304	355.501	359.801	862.5679	1222.3689	.32391	193.0
146	131.304	356.039	360.359	862.1740	1222.5330	.32592	191.9
147	132.304	356.574	360.913	861.7832	1222.6962	.32794	190.8
148	133.304	357.106	361.465	861.3934	1222.8584	.32995	189.7
149	134.304	357.635	362.013	861.0068	1223.0198	.33196	188.6
150	135.304	358.161	362.559	860.6213	1223.1803	.33400	187.5
151	136.304	358.683	363.100	860.2399	1223.3399	.33580	186.4
152	137.304	359.203	363.640	859.8588	1223.4988	.33761	185.3
153	138.304	359.721	364.177	859.4799	1223.6569	.33942	184.3
154	139.304	360.236	364.711	859.1031	1223.8141	.34123	183.3
155	140.304	360.749	365.243	858.7276	1223.9706	.34304	182.3
156	141.304	361.260	365.773	858.3533	1224.1263	.34485	181.3
157	142.304	361.768	366.300	857.9811	1224.2811	.34666	180.3
158	143.304	362.273	366.824	857.6112	1224.4352	.34847	179.3

TABLE No. 8.—CONTINUED.

The Properties of Saturated Steam.

Pressure per Square Inch.		Temperature in Fahrenheit Degrees.	Number of British Thermal Units in One Pound from Zero (Fahrenheit).			Weight of One Cubic Foot of Steam in Decimals of a Pound.	Number of Cubic Feet of Steam from One Cubic Foot of Water.
Total Pressure in Pounds from a Vacuum	Pressure in Pounds as Shown by Steam Gauge.		Number of Units of Heat in Water.	Number of Units of Heat Required for Evaporation, Called Latent Heat.	Total Number of Units of Heat Contained in Steam.		
159	144.304	362.776	367.347	857.2415	1224.5885	.35028	178.3
160	145.304	363.277	367.867	856.8740	1224.7410	.35209	177.3
161	146.304	363.774	368.383	856.5099	1224.8929	.35397	176.4
162	147.304	364.270	368.898	856.1461	1225.0441	.35585	175.5
163	148.304	364.764	369.410	855.7846	1225.1946	.35773	174.6
164	149.304	365.255	369.920	855.4243	1225.3443	.35961	173.7
165	150.304	365.744	370.428	855.0654	1225.4934	.36149	172.8
166	151.304	366.232	370.934	854.7077	1225.6417	.36337	171.9
167	152.304	366.717	371.438	854.3514	1225.7894	.36525	171.0
168	153.304	367.199	371.939	853.9974	1225.9364	.36714	170.1
169	154.304	367.680	372.437	853.6456	1226.0826	.36903	169.2
170	155.304	368.158	372.934	853.2942	1226.2282	.37092	168.4
171	156.304	368.632	373.427	852.9461	1226.3731	.37272	167.6
172	157.304	369.105	373.918	852.5995	1226.5175	.37452	166.8
173	158.304	369.576	374.408	852.2533	1226.6613	.37632	166.0
174	159.304	370.045	374.895	851.9094	1226.8044	.37812	165.2
175	160.304	370.512	375.380	851.5670	1226.9470	.37992	164.4
176	161.304	370.978	375.865	851.2239	1227.0889	.38172	163.6
177	162.304	371.442	376.347	850.8833	1227.2303	.38353	162.8
178	163.304	371.904	376.827	850.5441	1227.3711	.38534	162.0
179	164.304	372.364	377.305	850.2062	1227.5112	.38715	161.2
180	165.304	372.822	377.781	849.8698	1227.6508	.38895	160.4
181	166.304	373.275	378.255	849.5347	1227.7897	.39077	159.7
182	167.304	373.731	378.727	849.2011	1227.9281	.39259	159.0
183	168.304	374.183	379.197	848.8689	1228.0659	.39441	158.3
184	169.304	374.633	279.665	848.5380	1228.2030	.39624	157.6
185	170.304	375.081	380.131	848.2086	1228.3396	.39807	156.9
186	171.304	375.527	380.595	847.8805	1228.4755	.39990	156.2
187	172.304	375.971	381.056	847.5549	1228.6109	.40173	155.5
188	173.304	376.413	381.516	847.2297	1228.7457	.40356	154.8
189	174.304	376.853	381.974	846.9058	1228.8798	.40539	154.1
190	175.304	377.291	382.429	846.5844	1229.0134	.40722	153.4
191	176.304	377.727	382.883	846.2633	1229.1463	.40899	152.7
192	177.304	378.161	383.335	845.9437	1229.2787	.41076	152.0
193	178.304	378.593	383.785	845.6256	1229.4106	.41253	151.3
194	179.304	379.023	384.233	845.3089	1229.5419	.41430	150.7
195	180.304	379.452	384.679	844.9938	1229.6728	.41607	150.1
196	181.304	379.979	385.123	844.6801	1229.8031	.41784	149.5
197	182.304	380.305	385.567	844.3660	1229.9330	.41962	148.9
198	183.304	380.729	386.008	844.0543	1230.0623	.42140	148.3
199	184.304	381.152	386.449	843.7422	1230.1912	.42318	147.7
200	185.304	381.573	386.887	843.4326	1230.3196	.42496	147.1
201	186.304	381.992	387.324	843.1234	1230.4474	.42667	146.5
202	187.304	382.410	387.760	842.8148	1230.5748	.42838	145.9
203	188.304	382.827	388.194	842.5076	1230.7016	.43009	145.3
204	189.304	383.242	388.627	842.2010	1230.8280	.43180	144.7
205	190.304	383.655	389.057	841.8969	1230.9539	.43351	144.1
206	191.304	384.066	389.485	841.5942	1231.0792	.43523	143.5
207	192.304	384.475	389.912	841.2921	1231.2041	.43695	142.9
208	193.304	384.883	390.337	840.9914	1231.3284	.43866	142.3
209	194.304	385.288	390.759	840.6933	1231.4523	.44039	141.8
210	195.304	385.671	391.179	840.3967	1231.5757	.44211	141.3

TABLE No. 9.

The Properties of Water from 32° to 212° Fahrenheit.

Elastic Force.		Temperature in Fahrenheit Degrees.	Number of British Thermal Units in One Pound from Zero (Fahrenheit).			Weight of One Cubic Foot of Vapor in Decimals of a Pound.	Number of Cubic Feet of Steam from One Cubic Foot of Water.
In Pounds on the Square Inch.	In Inches of Mercury.		Number of Units of Heat in Water.	Number of Units of Heat Required for Evaporation, Called Latent Heat.	Total Number of Units of Heat Contained in Vapor.		
.089	.1811	32	32.000	1091.700	1123.700	.00030	208,080
.092	.1884	33	33.000	1091.005	1124.005	.00030	200,480
.096	.1960	34	34.000	1090.310	1124.310	.00031	193,180
.100	.2039	35	35.000	1089.615	1124.615	.00032	186,180
.104	.2121	36	36.000	1088.920	1124.920	.00033	179,380
.108	.2205	37	37.000	1088.225	1125.225	.00034	172,780
.112	.2292	38	38.000	1087.530	1125.530	.00036	166,380
.117	.2382	39	39.001	1086.834	1125.835	.00038	160,230
.122	.2476	40	40.001	1086.139	1126.140	.00040	154,330
.127	.2573	41	41.001	1085.444	1126.445	.00042	148,620
.132	.2673	42	42.001	1084.749	1126.750	.00043	143,220
.137	.2777	43	43.001	1084.054	1127.055	.00045	138,070
.142	.2884	44	44.002	1083.358	1127.360	.00047	133,120
.147	.2994	45	45.002	1082.663	1127.665	.00049	128,370
.152	.3109	46	46.002	1081.968	1127.970	.00050	123,840
.158	.3228	47	47.002	1081.273	1128.275	.00052	119,610
.164	.3351	48	48.003	1080.577	1128.580	.00054	115,490
.170	.3478	49	49.003	1079.882	1128.885	.00056	111,470
.176	.3608	50	50.003	1079.187	1129.190	.00058	107,630
.183	.3743	51	51.004	1078.491	1129.495	.00060	103,930
.190	.3883	52	52.004	1077.796	1129.800	.00062	100,330
.197	.4028	53	53.005	1077.100	1130.105	.00065	96,930
.205	.4177	54	54.005	1076.405	1130.410	.00067	93,680
.212	.4332	55	55.006	1075.709	1130.715	.00069	90,540
.220	.4492	56	56.006	1075.014	1131.020	.00071	87,500
.228	.4656	57	57.007	1074.318	1131.325	.00073	84,560
.236	.4825	58	58.007	1073.623	1131.630	.00076	81,740
.245	.5000	59	59.008	1072.927	1131.935	.00079	79,020
.254	.5180	60	60.009	1072.231	1132.240	.00082	76,370
.263	.5367	61	61.010	1071.535	1132.545	.00085	73,810
.273	.5560	62	62.011	1070.839	1132.850	.00088	71,330
.282	.5758	63	63.012	1070.143	1133.155	.00091	68,940
.292	.5962	64	64.013	1069.447	1133.460	.00094	66,630
.302	.6173	65	65.014	1068.751	1133.765	.00097	64,420
.313	.6391	66	66.015	1068.055	1134.070	.00100	62,290
.324	.6615	67	67.016	1067.359	1134.375	.00103	60,280
.335	.6846	68	68.018	1066.662	1134.680	.00107	58,340
.347	.7084	69	69.019	1065.966	1134.985	.00111	56,470
.359	.7330	70	70.020	1065.270	1135.290	.00115	54,660
.372	.7583	71	71.021	1064.574	1135.595	.00119	52,910
.385	.7844	72	72.023	1063.877	1135.900	.00123	51,210
.398	.8114	73	73.024	1063.181	1136.205	.00127	49,570
.411	.8391	74	74.026	1062.484	1136.510	.00131	48,000
.425	.8676	75	75.027	1061.788	1136.815	.00135	46,510
.440	.8969	76	76.029	1061.091	1137.120	.00139	45,060
.455	.9271	77	77.030	1060.395	1137.425	.00143	43,650
.470	.9583	78	78.032	1059.698	1137.730	.00148	42,280
.486	.9905	79	79.034	1059.001	1138.035	.00153	40,960
.502	1.023	80	80.036	1058.304	1138.340	.00158	39,690
.518	1.056	81	81.037	1057.608	1138.645	.00163	38,480
.535	1.091	82	82.039	1056.911	1138.950	.00168	37,320

TABLE No. 9.—CONTINUED.

The Properties of Water from 32° to 212° Fahrenheit.

Elastic Force.		Temperature in Fahrenheit Degrees.	Number of British Thermal Units in One Pound from Zero (Fahrenheit).			Weight of One Cubic Foot of Vapor in Decimals of a Pound.	Number of Cubic Feet of Steam from One Cubic Foot of Water.
In Pounds on the Square Inch.	In Inches of Mercury.		Number of Units of Heat in Water.	Number of Units of Heat Required for Evaporation, Called Latent Heat.	Total Number of Units of Heat Contained in Vapor.		
.553	1.127	83	83.041	1056.214	1139.255	.00173	36,190
.571	1.163	84	84.043	1055.517	1139.560	.00178	35,100
.590	1.201	85	85.045	1054.820	1139.865	.00183	34,050
.609	1:240	86	86.047	1054.123	1140.170	.00189	33,030
.629	1.281	87	87.049	1053.426	1140.475	.00195	32,050
.650	1.323	88	88.051	1052.729	1140.780	.00201	31,100
.671	1.366	89	89.053	1052.032	1141.085	.00207	30,180
.692	1.410	90	90.055	1051.335	1141.390	.00213	29,290
.715	1.454	91	91.057	1050.638	1141.695	.00219	28,430
.738	1.500	92	92.059	1049.941	1142.000	.00226	27,600
.761	1.548	93	93.061	1049.244	1142.305	.00233	26,800
.785	1.597	94	94.063	1048.547	1142.610	.00240	26,020
.809	1.647	95	95.065	1047.850	1142.915	.00247	25,270
.834	1.698	96	96.068	1047.152	1143.220	.00254	24,540
.860	1.751	97	97.071	1046.454	1143.525	.00262	23,830
.887	1.805	98	98.074	1045.756	1143.830	.00270	23,140
.914	1.861	99	99.077	1045.058	1144.135	.00278	22,470
.943	1.918	100	100.080	1044.360	1144.440	.00286	21,830
.972	1.977	101	101.083	1043.662	1144.745	.00294	21,210
1.001	2.037	102	102.086	1042.964	1145.050	.00302	20,620
1.031	2.099	103	103.089	1042.266	1145.355	.00311	20,050
1.062	2.163	104	104.092	1041.568	1145.660	.00320	19,500
1.094	2.227	105	105.095	1040.870	1145.965	.00330	18,970
1.126	2.293	106	106.098	1040.172	1146.270	.00340	18,460
1.159	2.361	107	107.101	1039.474	1146.575	.00350	17,960
1.193	2.431	108	108.104	1038.776	1146.880	.00360	17,470
1.229	2.503	109	109.107	1038.078	1147.185	.00370	16,990
1.265	2.577	110	110.110	1037.380	1147.490	.00380	16,520
1.302	2.653	111	111.113	1036.682	1147.795	.00390	16,070
1.341	2.731	112	112.117	1035.983	1148.100	.00400	15,640
1.381	2.810	113	113.121	1035.284	1148.405	.00410	15,220
1.421	2.892	114	114.125	1034.585	1148.710	.00421	14,820
1.462	2.976	115	115.129	1033.886	1149.015	.00433	14,430
1.504	3.061	116	116.133	1033.187	1149.320	.00445	14,050
1.547	3.149	117	117.137	1032.488	1149.625	.00457	13,680
1.591	3.239	118	118.141	1031.789	1149.930	.00470	13,320
1.636	3.331	119	119.145	1031.090	1150.235	.00483	12,970
1.682	3.425	120	120.149	1030.391	1150.540	.00496	12,630
1.730	3.522	121	121.153	1029.692	1150.845	.00508	12,300
1.779	3.621	122	122.157	1028.993	1151.150	.00521	11,980
1.828	3.723	123	123.161	1028.294	1151.455	.00535	11,670
1.879	3.826	124	124.165	1027.595	1151.760	.00549	11,370
1.931	3.933	125	125.169	1026.896	1152.065	.00563	11,080
1.984	4.042	126	126.173	1026.197	1152.370	.00578	10,800
2.039	4.153	127	127.177	1025.498	1152.675	.00593	10,530
2.096	4.267	128	128.182	1024.798	1152.980	.00608	10,265
2.154	4.384	129	129.187	1024.098	1153.285	.00624	10,010
2.213	4.503	130	130.192	1023.398	1153.590	.00640	9,760
2.273	4.625	131	131.197	1022.698	1153.895	.00656	9,516
2.335	4.750	132	132.202	1021.998	1154.200	.00673	9,276
2.398	4.878	133	133.207	1021.298	1154.505	.00690	9,046

TABLE NO 9.—CONTINUED.

The Properties of Water from 32° to 212° Fahrenheit.

ELASTIC FORCE.		Temperature in Fahrenheit Degrees.	NUMBER OF BRITISH THERMAL UNITS IN ONE POUND FROM ZERO (FAHRENHEIT).			Weight of One Cubic Foot of Vapor in Decimals of a Pound.	Number of Cubic Feet of Steam from One Cubic Foot of Water.
In Pounds on the Square Inch.	In Inches of Mercury.		Number of Units of Heat in Water.	Number of Units of Heat Required for Evaporation, Called Latent Heat.	Total Number of Units of Heat Contained in Vapor.		
2.461	5.009	134	134.212	1020.598	1154.810	.00707	8,826
2.526	5.143	135	135.217	1019.898	1155.115	.00725	8,611
2.594	5.280	136	136.222	1019.198	1155.420	.00743	8,401
2.663	5.420	137	137.227	1018.498	1155.725	.00761	8,191
2.732	5.563	138	138.223	1017.797	1156.030	.00780	7,991
2.803	5.709	139	139.239	1017.096	1156.335	.00799	7,798
2.876	5.858	140	140.245	1016.395	1156.640	.00819	7,613
2.952	6.011	141	141.251	1015.694	1156.945	.00839	7,433
3.029	6.167	142	142.257	1014.993	1157.250	.00860	7,258
3.108	6.327	143	143.263	1014.292	1157.555	.00881	7,088
3.188	6.490	144	144.269	1013.591	1157.860	.00903	6,920
3.270	6.657	145	145.275	1012.890	1158.165	.00925	6,755
3.353	6.827	146	146.281	1012.189	1158.470	.00948	6,595
3.438	7.001	147	147.287	1011.488	1158.775	.00971	6,440
3.526	7.179	148	148.293	1010.787	1159.080	.00993	6,290
3.615	7.361	149	149.299	1010.086	1159.385	.01016	6,144
3.707	7.547	150	150.305	1009.385	1159.690	.01040	6,004
3.801	7.736	151	151.311	1008.684	1159.995	.01064	5,867
3.896	7.929	152	152.318	1007.982	1160.300	.01089	5,734
3.992	8.127	153	153.325	1007.280	1160.605	.01114	5,604
4.090	8.329	154	154.332	1006.578	1160.910	.01140	5,477
4.191	8.535	155	155.339	1005.876	1161.215	.01167	5,353
4.295	8.745	156	156.346	1005.174	1161.520	.01194	5,232
4.400	8.959	157	157.353	1004.472	1161.825	.01222	5,114
4.507	9.178	158	158.360	1003.770	1162.130	.01250	5,000
4.617	9.401	159	159.367	1003.068	1162.435	.01279	4,888
4.729	9.629	160	160.374	1002.366	1162.740	.01308	4,779
4.843	9.861	161	161.381	1001.664	1163.045	.01338	4,673
4.960	10.098	162	162.389	1000.961	1163.350	.01368	4,569
5.079	10.340	163	163.397	1000.258	1163.655	.01399	4,467
5.200	10.588	164	164.405	999.555	1163.960	.01430	4,368
5.324	10.840	165	165.413	998.852	1164.265	.01462	4,271
5.451	11.097	166	166.421	998.149	1164.570	.01495	4,177
5.580	11.359	167	167.429	997.446	1164.875	.01528	4,085
5.711	11.627	168	168.437	996.743	1165.180	.01562	3,996
5.845	11.900	169	169.445	996.040	1165.485	.01596	3,910
5.981	12.178	170	170.453	995.337	1165.790	.01631	3,826
6.120	12.461	171	171.461	994.634	1166.095	.01667	3,744
6.262	12.750	172	172.470	993.930	1166.400	.01704	3,664
6.408	13.045	173	173.479	993.226	1166.705	.01741	3,586
6.555	13.345	174	174.488	992.522	1167.010	.01779	3,510
6.704	13.651	175	175.497	991.818	1167.315	.01817	3,436
6.857	13.963	176	176.506	991.114	1167.620	.01855	3,365
7.013	14.281	177	177.515	990.410	1167.925	.01894	3,295
7.172	14.605	178	178.524	989.706	1168.230	.01934	3,226
7.335	14.935	179	179.533	989.002	1168.535	.01975	3,159
7.500	15.271	180	180.542	988.298	1168.840	.02017	3,093
7.668	15.614	181	181.551	987.594	1169.145	.02060	3,029
7.841	15.963	182	182.561	986.889	1169.450	.02104	2,966
8.016	16.318	183	183.571	986.184	1169.755	.02148	2,905
8.194	16.680	184	184.581	985.479	1170.060	.02193	2,846

TABLE NO. 9.—CONTINUED.

The Properties of Water from 32° to 212° Fahrenheit.

Elastic Force.		Temperature in Fahrenheit Degrees.	Number of British Thermal Units in One Pound from Zero (Fahrenheit).			Weight of One Cubic Foot of Vapor in Decimals of a Pound.	Number of Cubic Feet of Steam from One Cubic Foot of Water.
In Pounds on the Square Inch.	In Inches of Mercury.		Number of Units of Heat in Water.	Number of Units of Heat Required for Evaporation, Called Latent Heat.	Total Number of Units of Heat Contained in Vapor.		
8.375	17.049	185	185.591	984.774	1170.365	.02238	2,789
8.558	17.425	186	186.601	984.069	1170.670	.02284	2,733
8.745	17.807	187	187.611	983.364	1170.975	.02331	2,678
8.936	18.196	188	188.621	982.659	1171.280	.02379	2,624
9.132	18.593	189	189.632	981.953	1171.585	.02428	2,571
9.330	18.997	190	190.643	981.247	1171.890	.02470	2,519
9.532	19.408	191	191.654	980.541	1172.195	.02529	2,469
9.738	19.827	192	192.665	979.835	1172.500	.02580	2,420
9.947	20.253	193	193.676	979.129	1172.805	.02632	2,372
10.160	20.687	194	194.686	978.424	1173.110	.02685	2,325
10.377	21.129	195	195.697	977.718	1173.415	.02740	2,279
10.597	21.579	196	196.708	977.012	1173.720	.02796	2,234
10.822	22.036	197	197.719	976.306	1174.025	.02853	2,190
11.051	22.502	198	198.730	975.600	1174.330	.02910	2,147
11.284	22.976	199	199.741	974.894	1174.635	.02967	2,105
11.521	23.458	200	200.753	974.187	1174.940	.03025	2,064
11.761	23.948	201	201.765	973.480	1175.245	.03083	2,024
12.006	24.446	202	202.777	972.773	1175.550	.03142	1,985
12.255	24.953	203	203.789	972.066	1175.855	.03201	1,953
12.508	25.468	204	204.801	971.359	1176.160	.03261	1,916
12.766	25.992	205	205.813	970.652	1176.465	.03323	1,880
13.028	26.525	206	206.825	969.945	1176.770	.03386	1,844
13.295	27.067	207	207.837	969.238	1177.075	.03450	1,809
13.568	27.619	208	208.849	968.531	1177.380	.03516	1,775
13.843	28.180	209	209.861	967.824	1177.685	.03584	1,741
14.122	28.751	210	210.874	967.116	1177.990	.03654	1,708
14.406	29.332	211	211.887	966.408	1178.295	.03725	1,676
14.700	29.9218	212	212.900	965.700	1178.600	.03797	1,644

INDEX.

PAGE

A

Absolute information from diagram, - 79
" pressure, - - - - - - 21
Action of steam in cylinder, theory of - 137
" " " " jacketed cylinders, - 141
Actual curves, - - - - - - - 105
Adiabatic curve, - - - - - - - 105
Adjuster for indicator cord, - - 31-72
Admission line, - - - - - - - 92
" " late, - - - - - - 150
" " correctness of - - - 148
" " method of determining the correctness of, 149-151
Amsler polar planimeter, - - - - - 198
" " " mean effective pressure from, 198
Application of indicator, - - - - - 35
Areas and circumferences of circles, 289
Areas of large circles, finding 294
Atmospheric line, - - - - - 75-91
" " tracing, - - - 75
Averager for measuring diagrams, - - 185
Available power, - - - - - - 221

B

Back pressure, - - - - - - - 219
" " cause of excessive, - - 219
" " in condensing and non-condensing engines, 220-227
" " loss from, - - - - - 219
" " total - - - - - - 22
" " line, - - - - - 22-94
Blanks, example of printed, - - - 78
Boiler pressure, - - - - - - - 21
Boilers, testing, - - - - - - - 277

C

Calculating water consumption by constant, - - - - - - - - 165
Care and use of indicator, - - - 68
Carrying pulleys, - - - - - - 71
Calorimeter tests, manner of making, - 248
" " rule for computing, - 251

PAGE

Calorimeters, different kinds, - - - 244
Calorimeter, description of barrel, - 248
" Prof. Carpenter's, - - 255
Clearance, - - - - - - 22-23-87
" estimating, - - - - - 88
" per cent. - - - - - - 23
" the effect of - - - - 219
" line, - - - - - - - 87
" " locating, - - - - 88
Close agreement of the actual with the isothermal in jacketed cylinders, 143
Coffin averager, - - - - - - 185
" " construction of, - - 186
" " finding the mean effective pressure with - 188
" " principle of operation of the - - - 190-197
Compression, - - - - - - - 22
" and clearance, - - . 219
" curve, - - - - - - 95
Comparison of diagrams from throttling and cut-off engines, - - - 201
Comparison of results between large and small clearance, - - - - 227
Condensing engine diagrams, - 266-267
Condensing and non-condensing diagrams compared, - - - - - 227
Condensation in cylinders, - - - 138
" loss from - - - - 142
Condenser, effect of adding - - - 214
" pressure in - - - - 214
Condensation in unjacketed cylinders, - 139
Cock, three way - - - - - - 30
" section of three way - - - 31
Cord, indicator, - - - - - - 72
" adjuster - - - - - - - 31
" stretching of the - - - - 72
Computing horse power, - - - - 112
Computing steam accounted for by the indicator, different methods of 152-157
Combining diagrams, method of - 229-233
" " from compound engines, - - 228

PAGE

Compound condensing engines, diagrams from, - - - - 261-268
Curves, theoretical or isothermal - - 97
" agreement of actual and theoretical expansion, - - - 106
Cut-off, the point of - - - - 93-213
" locating the point of - - 108-111
" the best point to - - - - 214

D

Data necessary, - - - - - - 77
" how to preserve - - - - 77
Description of indicator, - - - - 16
Device, indicator testing, - - - - 175
Diagrams, indicator - - - - - 16
" the information to be derived from - - - - 79-82
" from Marsh steam pump, - 274
" combining, - - - - 228
Diagram, essential features of a well-formed - - - - - 84
" lines and points of the - - 91
Diagrams, to take simultaneously - - 63
" the outline of - - - - 83
" engine underloaded - - 117
" not desirable - - - - 123
" reading the - - - - 146
Diagram representing the economy of high pressure - - - - 211
Diagram showing the most favorable point of cut-off - - - - 216
Diagram from a steam jacketed cylinder, - - - - - - - - 143
Diagrams from both ends of a cylinder, 85
Diagram, good features of a - - - 80
Diagrams, measuring a large number of 185
Diagram, names of the lines of the 91-95
Diagrams, study of - - - - - 83
" method of taking - - - 75
" from pumps, - - - 272-273
Diagram averager - - - - - . 185
Diagrams from various engines, - 258-267
Diagrams to show the effect between large and small clearance, - 219 226
Diagrams from steam cylinder and ammonia compressor, - - 239-240
Diagrams from gas engines, - - - 241
" from oil engines, - - - 242
" miscellaneous, - - 258-267
Directions for indicating, - - - 75
Driving gear for indicator, - - - 54
Drum motion, testing accuracy of - - 52
Drum stop motion, - - - - 61-62
Drum spring, - - - - - - - 74

PAGE

E

Economy in the use of steam - - - 282
" of expansion, - - 123-125-208
" " heating feed water, - - 284
" " high pressure, - - - 209
" steam engine - - - - 275
Engines, testing - - - - - 282
" compound - - - - - 267
" comparison of diagrams from throttling and cut-off - 202-207
Essential features of diagrams, - - 84
Exhaust-closure, - - - - - - 94
Exhaust-closure, early and late, - - 219
Exhaust, heat lost at - - - - 210
Exhaust-line, - - - - - - - 93
Exhaust-valve, leaky - - - - 148
Expansion of steam, - - - - 119
" line or curve, - - - - 93
" ratio of - - - - - 130
" initial - - - - - 22
" curve, peculiarities of - - 83
" line, and leaky valves, - - 147
" curve, agreement of actual and theoretical - 143-145
" curve, geometrical method of finding points in the - 101
Experiments to determine the most favorable point of cut-off, - - 215

F

Feed-water, economy of heating - - 284
Feed-water accounted for by the indicator, - - - - - - 152
Foot pound, - - - - - - 112
Friction, effects of - - - - - 179
" of indicator piston - - - 181
" " " " how to determine - - - - - 182

G

Gases, Mariotte law concerning - - 97
Gas engines, diagrams from - - 240-241
Gauge or boiler pressure, - - - 21
Guide or carrying pulleys, - - - 71
Guide pulleys, disadvantages of - - 71
Gas and oil engine diagrams - - - 234

H

Heat lost, - - - - - - - 214
" available - - - - - - 210
" latent - - - - - - 20
" lost at exhaust - - - - - 210
" sensible - - - - - - 20
" the unit of - - - - - - 20

PAGE

Heat units - - - - - - 20
High pressure, economy of - - - 209
High speed diagram compared with moderate speed, - - - 267-268
High speed engines, manner of using the cord on - - - - - 51
Holes in cylinder, position for indicator, - - - - - - - 36-37
Hooking the cords, - - - - - - 51
Horse power, elements of - - - 112
" " finding the - - - 112
" " for one pound mean effective pressure, - - 113
" " indicated - - - - - 21
" " net - - - - - - 21
" " factors for - - - - - 113
" " by ordinates - - - 114
Hyperbolic logarithms, - - - - 128
" curve - - - - - 97
" logarithms, table of - - 129

I

Indicator, brief history of the - - 11
" Watts' original - - - 11
" " " construction of 11
" McNaught - - - - 12
" " description of - 13
" parts of - - - - - - 14
" size of piston of - - - 14
" purpose of the - - - 16
" care required in the use of - 68
" Tabor - - - - - 24-25
" construction of parts of - 24-30
" oiling the - - - - 68-183
" scales - - - - 73-87
" springs - - - - 32-33-34
" " to use - - - - - 73
" appliances, cuts of - - 44-53
" electrical appliance - - 65
" testing devise, - - - 175
" " " construction of 176
" " " manner of using 176
" " " information derived from - 180
" reducing gear, - - - 54
" " " parts of - 55-57
" " " directions for using - 58-60
" cord - - - - - - - 72
" " adjuster - - - - - 31
" driving gear - - - - 54
" how to attach the - - 75
" piston, the effects of momentum of - - - - - 181

PAGE

Indicators, using one or two - - 30-77
Indicator diagrams, the production of 17-86
" " explanation of 17-18-19
" cylinder, warming up - - 75
" card, proportionate length of 38
" diagrams from various engines 257
" piston, effects of a tightly fitting - - - - - 182
Initial expansion, - - - - - 95
Initial pressure, - - - - 92-213
Isothermal curve, - - - - - 97
" " locating points on the 98
" " geometrical method for constructing the 101

J

Jacketing, cylinder, - - - - 143

L

Latent heat, - - - - 20
Lazy tongs, - - - 39-40
" " proportion of - - 40-41-42
Leakage - - - - - 79
Leaky Valves, detecting by the indicator 106
Length of diagrams, - - - 38
Levers, reducing - - - 46-49
Locating clearance line on diagram, 88
" point of cut-off - - 108-111
Long indicator pipes, - - - 37

M

Management and care of indicator, - 68
Mariotte law of gases, - - - 97
Manner of taking diagrams, - - 75
Marsh steam pump diagrams, - - 274
Mean effective pressure, - - 22-95
" " " computing - 117
" " " measuring by ordinates the 114
Mean effective pressure, horse power for one pound, - - - 113
Mean pressure of expanding steam, the rule for finding the - - 130-131
Measuring diagrams, - - - 114
Memoranda of diagrams, making - 77
Method of indicating a steam engine - 75
" " computing horse power, 113-114
Moisture in steam, - - - - 214
Motion, Watts' parallel - - - 14
" straight line - - - 15
" drum - - - 16-70
Movement of pencil, - - - 16

PAGE

O

Oil engines, operation of - - - 243
Oil engine diagrams, - - - 242
Oiling the indicator, - - 68-183
Ordinates, horse power by - - 114
" different methods of measuring the - - 116-117

P

Paper-drum, - - - - 16
Pantagraph - - - 44-45
" adjusting the - - - 45
Parallel motion, Watts . - - 14
Parallel or straight line motion, - - 15
Pencil, adjusting the - - 69-70
Pencil movement, proportioning of 35-36
Pendulum reducing gear, - - 47-49
Piping for indicators, - - - 36
Planimeters, - - - - 184
" Amsler polar - - 198
Planimeter, measuring mean effective pressure by the - - - 184
Point of cut-off, - - - 93-213
" " exhaust, - - - - 93
Pressure, absolute - - - 21-22
" back - - - - 219
" boiler - - - - 21
" initial - - - - 21
" mean effective - - 22-95
" of expanding steam - - 123
" of expanding steam, rule for finding the - - 130-131
" terminal - - - - 22
" total back - - - 22
" for indicator springs - - 73
Priming, extent of - - - 244
Printed blanks, - - - - 78
Pulleys, carrying - - - - 71
Pump diagrams, - - - 272-273

R

Ratio of expansion, - - - 130
Reducing levers, - - - - 47
Reducing gears, different kinds of 50-53
Re-evaporation, - - - - 138
Resistance to motion of the piston of a steam engine, - - - 219
Rule for finding the mean pressure of expanding steam, - - 130-131

S

Safe pressure for indicator springs, - 73
Saturated steam, - - - - 21

PAGE

Scales, indicator - - - - 73
Sensible heat, - - - - 20
Serrated lines of the diagram, - - 181
Specific heat, - - - - 20
Springs, how marked, - - - 73
Spring to use, proper - - - 73
Springfield gas engine diagrams, 237-238
Steam, dry - - - - - 21
" expansion of - - - 119
" line - - - - - 92
" exhausted per hour, - - 152
" line, straight - - - 209
" per horse power per hour - 152
" jacketed cylinders, diagrams from - - - - 143
" pressure of expanding - - 123
" pipe, large and small, - - 79
" pressure, working with high and low, - - - - 208
" properties of saturated - - 295
" saved in clearance, - - 160
" superheated, - - - 21
" or water consumption, - - 23
" admission of - - - 84
" connections for indicators, - 35
" in cylinders, (action of) - - 137
" accounted for by the indicator, - 152
" valve, leaky - - - 79

T

Tabor indicator, - - - 24-25
" " with reducing motion and parts, - 54-57
" " with electrical attachment - - - 65
" " with combined pistons, 235
Table of hyperbolic logarithms - - 129
" showing the theoretical economy of using steam expansively, - 132
" of constants for finding the average pressure with any pressure of steam, - - - - 134
" of average pressure of steam in the cylinder with different rates of expansion, - - - - 135
" quantity of steam accounted for by the indicator, - - 170-171
" showing saving effected by the use of feed-water heaters, - - 286
" areas and circumferences of circles 289
" properties of saturated steam, - 295
" properties of water, - - 299
Tecnnical terms - - 20-21-22-23
Terminal pressure, - - 95-147-275

PAGE

Terminal pressure, cause of high - 147
" " theoretical - - 147
Tests, different methods of making - 277
" what should be noted in making, 277
Testing engines and boilers, - - 279
Theoretical curve, object of drawing the 102
" " points from which to draw the - - 98
" " reasons for establishing the - - 103
Theoretical diagram, - - - 120
" economy of expansion, 126–133
Three-way cock, - - - - 30

U

Underloaded engines, - - 116–212
Unit of heat, - - - - 20
" " work, - - - - 20
Uses to which the indicator may be applied, - - - - 70

PAGE

V

Vacuum-line, establishing - - 87
Valves, adjusting - - - - 80
" detecting leaky - - - 79
Valve lap, - - - - - 23
" lead, - - - - - 23
" gear, incorrect, - - - 82

W

Watts' original indicator, - - - 11
" parallel motion, - - - 14
Water consumption, making allowance for - - - - - 164
" per horse power per hour - 152
Wire drawing - - - - 23
Work and heat - - - - 210
" in the two ends of the cylinder - 83
" the unit of - - - - 20

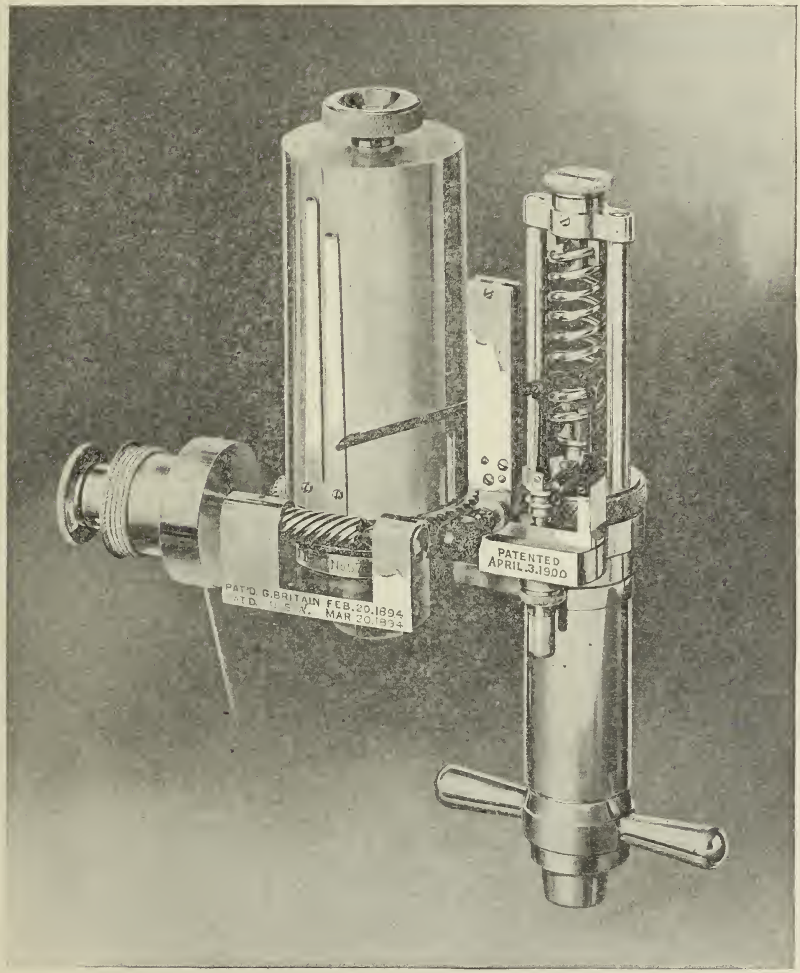
Z

Zero line, - - - - - 86

ADVERTISEMENTS.

THE IMPROVED TABOR STEAM ENGINE INDICATOR

With Outside Spring.



With Houghtaling Reducing Motion.

Coffin Averagers, Amsler Planimeters, Pantographs, Lazy Tongs, Ashcroft Reducing Wheels, Three Way Cocks, Carrying Pulleys, Etc.

Awarded Grand Prize at World's Fair, St. Louis, 1904.

THE ASHCROFT MANUFACTURING CO.,

SOLE MANUFACTURERS.

85-87-89 LIBERTY STREET,
NEW YORK.

22-24-26 S. CANAL STREET,
CHICAGO.